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BUCKLING TESTS OF A 10-FOOT DIAMETER STIFFENED CYLINDER WITH RECTANGULAR CUTOUTS

(NASA-TM-88996) EUCKLING TESIS CF A 10 FOOT DIAMETER STIFFENED CYIINDER WITH RECTANGULAR CUTOUTS (NASA) 38 p Avail: NTIS HC A03/MF A01 CSCL 20K

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SUMMARY

A 10-foot diameter aluminum cylinder with rectangular cutouts in its ring and stringer stiffened wall was loaded to buckling failure by an end bending moment. A 24x27 inch cutout, centered on the compression side of the shell, was first cut into the cylinder. After testing, the cutout area was enlarged to a 36x36 inch square cutout that removed the material damaged by buckling during the first test. After the second buckling test, the cutout area was patched with an equivalent stiffness plate bolted over the cutout hole. The cylinder was rotated 120° and a 18x18 inch square hole cut into the center of the new compression side of the shell. Test specimen details, test procedures, and test results are presented.

INTRODUCTION

Analytical prebuckling solutions (refs. 1-8) show that the presence of a cutout has a pronounced effect on the stresses and displacements in a cylindrical shell under various loading conditions. Analytical solutions for the buckling loads of such cylinders, that adequately account for the asymmetric effect of the cutout on the shell, are difficult to obtain. Large cutouts produce nonlinearities that are very difficult for analytical solutions to predict. As a result, the design of cylindrical shells with cutouts has generally relied on experimental results and a few analytical solutions.

The testing of the cylindrical shell in this report was undertaken to provide test data for a class of cylindrical shells that has few published results. An existing 10-foot diameter integrally stiffened aluminum cylinder from a previously completed test program was obtained from the Marshall Space Flight Center for buckling tests in the Langley Structures Laboratory. The cylinder provided an inexpensive test specimen for yielding valuable buckling data for large ring and stringer stiffened cylindrical shells with large cutouts.

Rudimentary analysis of the shell, using the results in reference 5, indicated that the cylinder could be loaded to buckling with an end bending moment from the large scale bending machine in the Langley Structures Laboratory. By covering the shell cutout area with an equivalent extensional stiffness patch, the cylinder could be retested with various cutout sizes. In this manner, replicate test results could be obtained for different size cutouts for comparison purposes.

The object of this report is to provide the physical properties of the cylinder and the cutouts and to report the experimental results from the buckling tests. The results are intended to supply needed additional data base experience for design as discussed in reference 5.

EXPERIMENTAL PROCEDURE

Test Specimen

The cylindrical shell construction (see figs. 1 and 2) has an integrally ring-and-stringer stiffened wall. The shell wall geometry was mill machined into a flat aluminum plate and the plate roll formed to the desired radius. Three such rolled plates were welded together with three weld joints equally spaced around the circumference of the completed cylindrical shell.

The outside diameter of the shell was measured to be 120.66 inches on the average. The shell wall thickness was measured to be 0.10 inch. These dimensions and the average measured stiffener dimensions are given in figure 1. The overall shell length was measured to be 94.125 inches. The results of an imperfection survey of the shell are given in reference 9. To provide a transfer of the applied load to the outside of the cylinder, a large steel ring was machined and bolted to the end of the shell (fig. 2). In addition, individual steel backup plates (fig. 3) were bolted to the inside of the shell wall to aid in transfer of the applied load into the shell wall.

Cutouts

The position of the cutouts in the three-segment shell were centered relative to the weld joints and to the ends of the cylinder. The shell wall material was cut flush with the sides of the rings and stringers bounding the cutout (fig. 3). In this manner, two of the shell wall cutout sides were stringer supported and two were ring supported. The first cutout was 24x27 inches and centered in one of the three shell wall segments. The cutout area was delineated by eight stringer spacings (24 inches) and by six ring spacings (27 inches). The second cutout was a 36x36 inch opening made by enlarging the first cutout area and covered an area of eight rings by 12 stringers. (The strain gage results and visual examination of the shell wall from the 24x27 inch cutout test revealed that the permanent-set damage to the shell was confined to a local area close to the cutout. It was determined that a 36x36 inch cutout would safely remove all of the yielded wall material resulting from the first cutout test.) The third cutout was an 18x18 inch hole made in the center of another of the three shell wall segments. The 36x36 inch cutout opening was covered by an equivalent extensional-stiffness patch. The stiffness of the shell wall in the axial direction was computed and used to determine the thickness for the aluminum patch plate. The patch plate was then rolled to a radius to match the outside wall of the shell. The dimensions of the patch plate were made two stiffener spacings wider than the cut-out opening all around to allow room for bolting the patch to the shell wall.

Shell Material

A series of compression tests made on coupon specimens machined from the cutout material determined the physical properties of the aluminum shell wall material. The average elastic modulus is 10,700,000 psi, the Poisson ratio is 0.33, and the 2% offset yield stress was 59,000 psi. The scatter in the coupon test results were less than 0.5%.

TEST PROCEDURE

Load Application

The bending moment was applied to the shell by a large scale bending machine acting through a pair of steel loading rings attached to the outside of the shell wall (fig. 2). Steel clamping plates, figure 3, were used on the inside of the shell to assist the transfer of load from the loading rings to the shell wall. The loading rings were in turn bolted to a pair of steel conical shaped loading heads that can be seen in figure 4. One conical loading head was bolted to a rigid wall mounted to the laboratory floor (commonly called the "backstop"). The other conical loading head was attached to a steel end plate (fig. 5). The end plate was attached to a triangular loading frame through two pin-end loading arms.

The right-angled triangular loading frame of the large-scale bending machine had a main pivot through its vertical leg that was supported by two uprights from the laboratory floor (fig. 5). The triangular loading frame was thus free to rotate about the main pivot pin in the uprights. One vertex of the triangular loading frame was loaded upward by a hydraulic jack resting on the laboratory floor. The displacement type loading on the triangular loading frame from the hydraulic jack was reacted by the two uprights which allowed the triangular loading frame to rotate. The other two vertices of the triangular loading frame were attached to two pin-ended loading arms which were attached to the loading plate. Rotation of the triangular loading frame about the main pivot pin through the uprights from the action of the hydraulic jack caused the upper loading arm to push against the top part of the loading plate while the lower arm pulled on the bottom part. Thus, the pin-ended loading arms applied equal and opposite loads to the loading plate, producing a bending moment on the conical shape loading heads. The pinned end of the loading arms allowed the loaded end of the cylinder to rotate as well as translate upward relative to the fixed end of the cylinder attached to the backstop.

Instrumentation

Back-to-back strain gages were located around the edges of the cutouts. Each gage was centered relative to the stiffener cross-section as depicted in figure 6. Deflections at select locations around the top generator line of the shell wall, as well as around the cutout edges, were recorded using Linear Variable Displacement Transducers (LVDT's). Eleven LVDT's were mounted inside the shell along its center line on a special instrumentation beam cantilevered from the backstop (fig. 4). The LVDT's measured normal deflections of the cutout edges relative to the undeformed center line of the shell. Four of these LVDT's were centered on the ring and stringer intersections at each corner of the cutouts. Four LVDT's were located on the four edges half way between the corners, centered on the ring and stringer intersection at midspan of the cutout. Two LVDT's were placed nine inches away from the cutout edge along the shell wall generator. Finally, one LVDT was placed at the loaded end of the top shell wall generator line. The load, the strain gage responses, and the LVDT displacement data were recorded on magnetic tape using an automated data acquisition system.

RESULTS AND DISCUSSION

18x18 Inch Cutout

The shell with the 18x18 inch cutout, loaded to failure by an end bending moment, failed suddenly with a loud noise and a large loss in bending stiffness at a bending moment of 25,500,000 in-lbs (Table I). The displacements from the LVDT's for this test are shown in figures 7 and 8 and the position of each LVDT is marked with a numeral on the inset. The corresponding response curve is marked with the same numeral. The displacements along the top generator line of the cylinder shell wall are nonlinear as can be seen in figure 7. From figure 7, note that the top of the shell wall at the loaded end of the cylinder moved almost linearly, but the rest of the shell wall moved non-linearly inward toward the centerline of the shell (a phenomenon also noted in ref. 6). As can be seen in figure 8, the load-displacement responses were nonlinear around the edges of the cutout. The edges of the cutout area all began to move inward above 6,000,000 in-lbs. Approaching the failure load, the deflection at locations 9, 10, and 11 on the right side of the cutout (facing the back-stop) reversed direction and began to buckle outward at about 25,000,000 in-lbs. The deflections at locations 6, 7, and 8 on the left side of the other hand began tp buckle inward. The shell finally failed in an asymmetric mode with the right side buckled out and the left side buckled inward, as can be seen in the photograph in figure 9.

The back-to-back strain gage responses from 32 gages and their 16 locations around the 18x18 inch cutout are shown in figures 10 through 13. Figures 10 and 12 show the locations of the strain gages, which are all centered halfway between stiffener centerlines. The longitudinal strains adjacent to the cutout area at longitudinal positions A and D, figure 11, showed a very nearly linear response with applied bending moment. Bending of the edge stringers near the center of the cutout, as evidenced by the separation in the back-to-back gage response curves at longitudinal positions B and C, became prominent above an applied moment of 10,000,000 in-lbs, resulting in a strain reversal which occurred at 20,800,000 in-lbs. Upon examination of the inside-outside response curves, it was seen that the right side of the cutout was bending outward, while the left side is bending inward. The strain reversal at 20,800,000 in-lbs precedes the deflection reversal shown in figure 8 and the shell buckling at 25,500,000 in-lbs.

The circumferential strains for the 18x18 inch cutout, as shown in figure 13, are symmetric about the midspan of the cutout front and back edges. The corner areas at circumferential positions E and H adjacent to the cutout showed very little circumferential strain present. At the circumferential positions F and G near the midspan of the cutout edge, the back-to-back gage responses showed tensile strains with bending of the ring at the edge of the cutout, taking place from the onset of loading. In addition, these results showed that the front and back edges of the cutout were bending inward.

24x27 Inch Cutout

The shell with the 24x27 inch cutout failed suddenly with a loud noise and a sudden loss in bending stiffness at an applied bending moment of 22,600,000 in-lb (Table I). The test displacement results are shown in figures 14 and 15. The top of the shell wall at the loaded end of the cylinder moved nearly linearly but the cutout area began to move nonlinearly inward relative to the unloaded position from the onset of loading, as can be seen from figure 14. The deflections around the edge of the cutout, shown in figure 15, showed that the right side of the cutout (facing the backstop) reversed its inward movement near 22,000,000 in-lbs and began to buckle outward. The left side on the other hand, began to buckle inward. The cutout area buckled asymmetrically, as can be seen figure 16, with the right side bowed outward and the left side buckled inward.

The back-to-back strain gage response curves from 44 gage and their 20 locations around the 24x27 inch cutout are shown in figures 17 through 20. As can be seen from figure 18, the longitudinal strains were symmetric about the midspan of the edges of the cutout area. The longitudinal strains at positions A and F, just out from the corner,showed a high degree of linearity and very little bending. Bending of the right and left edge stringers between the midspan and corners, as evidenced by the separation in the back-to-back strain gage response curves at positions B through E in figure 18, began early with applied loading, resulting in a strain reversal at 21,100,000 in-lbs. The longitudinal strain responses in general appear to be more linear than the displacement curves in figures 14 and 15 indicate. An examination of the inside-outside response curves from right side to left side of the cutout in figure 18 revealed that the right side of the cutout was bending outward and the left side inward as indicated by the displacement results in figure 15.

The circumferential strains for the 24x27 inch cutout, as can be seen in figure 20, exhibited symmetry about the midspan of the front and back edges. The corner gages at circumferential positions G and K, figure 19, showed very little strain while the gages at circumferential positions H through J nearer to the middle span show some tensile strain in addition to bending of the ring on the cutout edge. The bending of the edge stiffener began to take place about half-way through the loading. The inside-outside strain gages responses show that the center of the front and rear edges were bending inward.

36x36 Inch Cutout

The shell with the 36x36 inch cutout failed suddenly, with a subdued noise and a sudden loss in bending stiffness at a bending moment of 17,500,000 in-lb Table I). The test displacements are shown in figures 21 and 22. The top of the shell wall at the loaded end of the cylinder moved nearly linearly, but the top of the shell wall on an axial, through the cutout area, began to move inward nonlinearly immediately upon loading as can be seen in figure 21. As can be seen in figure 22, the corners all moved outward while the center of the sides moved outward, substantially more than the corners. The deflections around the edge of the cutout showed a symmetric mode developing with load. Both the right and left sides (facing the backstop) deflected outward with increasing outward motion near failure. There were no deflection reversals for the 36x36 inch cutout. Unlike the two smaller cutout tests, all four corners of the 36x36 cutout area moved outward substantially. The midspan of the front and back edges showed a substantial inward displacement relative to the other displacements. The buckle shape which was symmetric around the cutout can be seen in the photograph in figure 23.

The back-to-back strain gage responses from 24 gages and their 12 locations around the 36x36 inch cutout are shown in figures 24 through 27. As can be seen from figure 25, the longitudinal strains were symmetrical along the cutout edges with respect to the longitudinal direction. Bending of the edge stringer, as evidenced by the separation in the back-to-back strain gage response curves, at longitudinal positions A through C, figure 24, began early with loading, resulting in a strain reversal near 11,400,000 in-lbs. An examination of the inside-outside strain gage response curves from left side to right side confirm the symmetric buckling mode for the cutout area depicted by the displacement curves in figure 22.

In figure 27, the circumferential strains near the corners of the 36x36 inch cutout area at circumferential positions D and F, figure 26, remained nearly zero as the loading started but exhibited tensile and bending strains that developed as the shell neared collapse. The circumferential strains exhibited a symmetric bending of the cutout edges for both the right side and the left side of the cutout. At the circumferential position E near the midspan, the circumferential strains were tensile with stiffener bending occurring from the onset of loading. The circumferential strain pattern showed that the right side of the front and back edges were bending inward while the left side was bending outward.

CONCLUSIONS

A 120.66-inch diameter by 94.125 inch long integrally ring and stringer stiffened cylinder has been tested in bending with three sizes of rectangular cutouts in the compression side of the shell wall. Failure occurred by buckling of the shell wall around the cutout in all three cases. Buckling modes for the 18x18 inch and the 24x17 inch cutouts were asymmetric about the cutout axes, however, the 36x36 inch cutout buckling mode was symmetric about the cutout axes. The buckling deflections about the cutouts showed nonlinear behavior confirmed by bending of the cutout edges as shown by the back-to-back strain gage response curves. Local bending of the cutout edges caused strain reversal to occur at applied bending moments of 20,800,000, 21,100,000, and 11,400,000 in-lbs for the 18x18, 24x27, and 36x36 inch cutouts respectively. The deflection curves for the 18x18 and 24x27 inch cutout test showed that the sides of the cutout area initially moved inward, but then the right side deflections reversed near collapse, resulting in an asymmetric buckle mode with respect to the circumferential direction for the cutout area. The 36x36 cutout area, however, moved outward from the onset of loading to collapse with no deflection reversal and with a symmetric mode pattern with respect to the circumferential direction. The buckling failure loads for the three tests were

. sily discerned by the sudden loss in bending stiffness and an accompanying loud noise. The failure loads were 25,500,000, 22,600,000, and 17,500,000 in-lbs for the 18x18, 24x27, and 36x36 inch cutouts, respectively.

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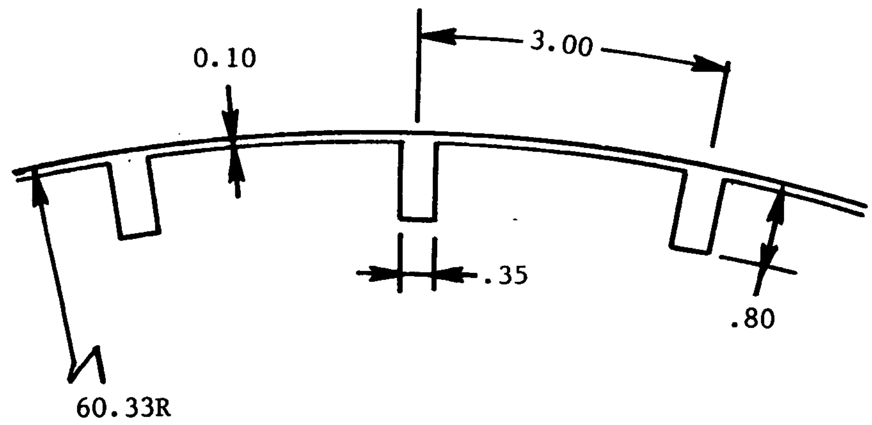
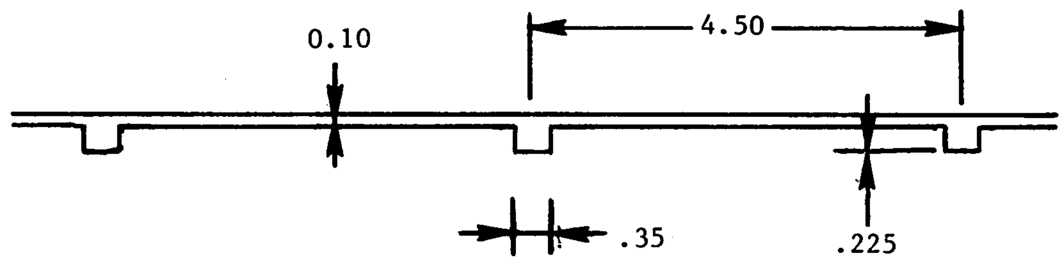
TABLE I. - EXPERIMENTAL BUCKLING RESULTS

Cutout Size (in x in)	Buckling Moment (in-lbs)	Cutout Area Parameter $\sqrt{\frac{r^2}{Rt}}$ *
18 x 18	25,500,000	2.64
24 x 27	22,600,000	3.74
36 x 36	17,500,000	5.29

* R is the shell wall radius, t is the shell wall equivalent thickness, r is the cutout area characteristic dimension (see ref. 5)



(a) Stringer details



(b) Ring details

Figure 1.- Geometric details of the shell wall (dimensions in inches).

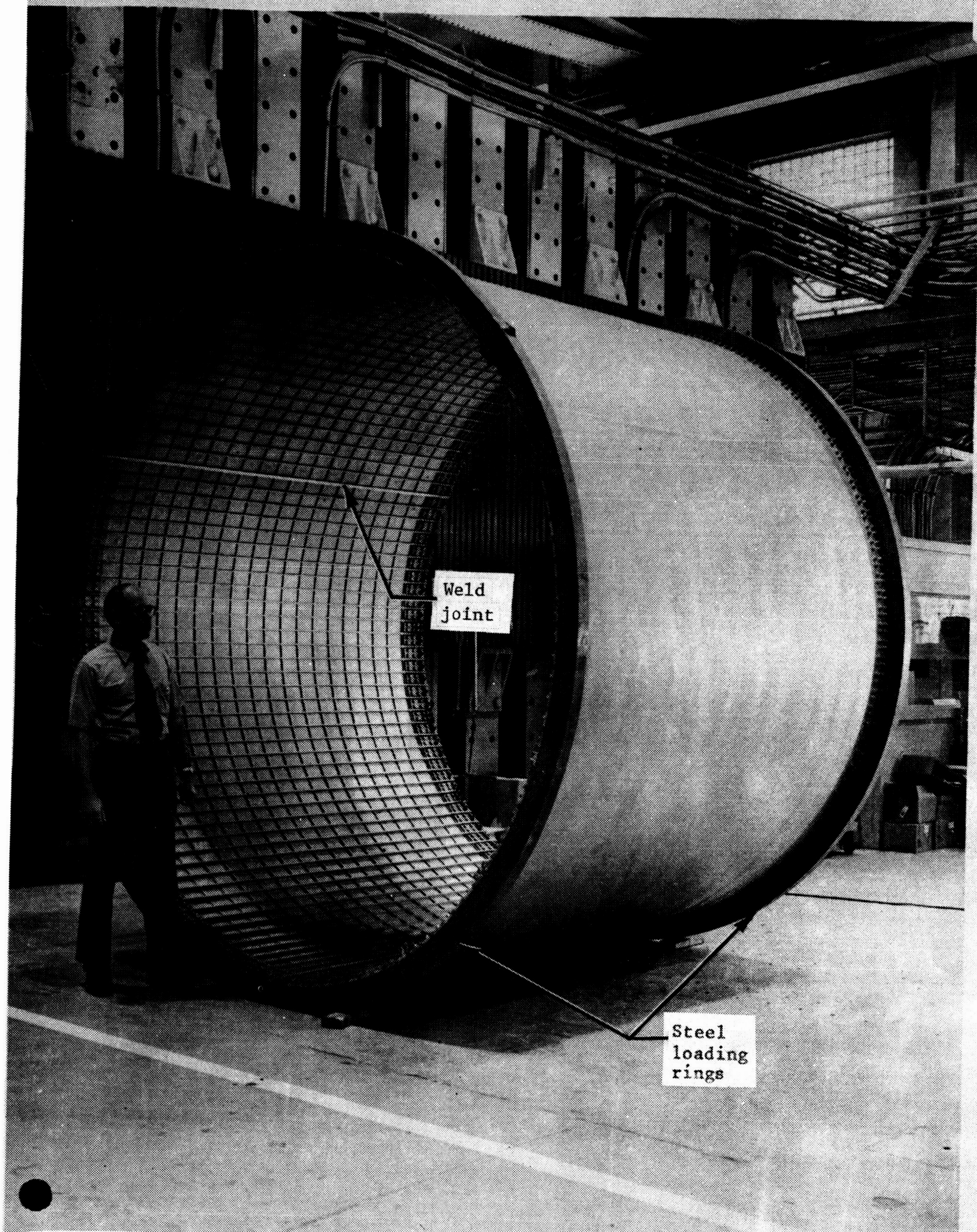


Figure 2.- The 10-foot diameter test cylinder with the steel loading rings attached to the ends of the shell.

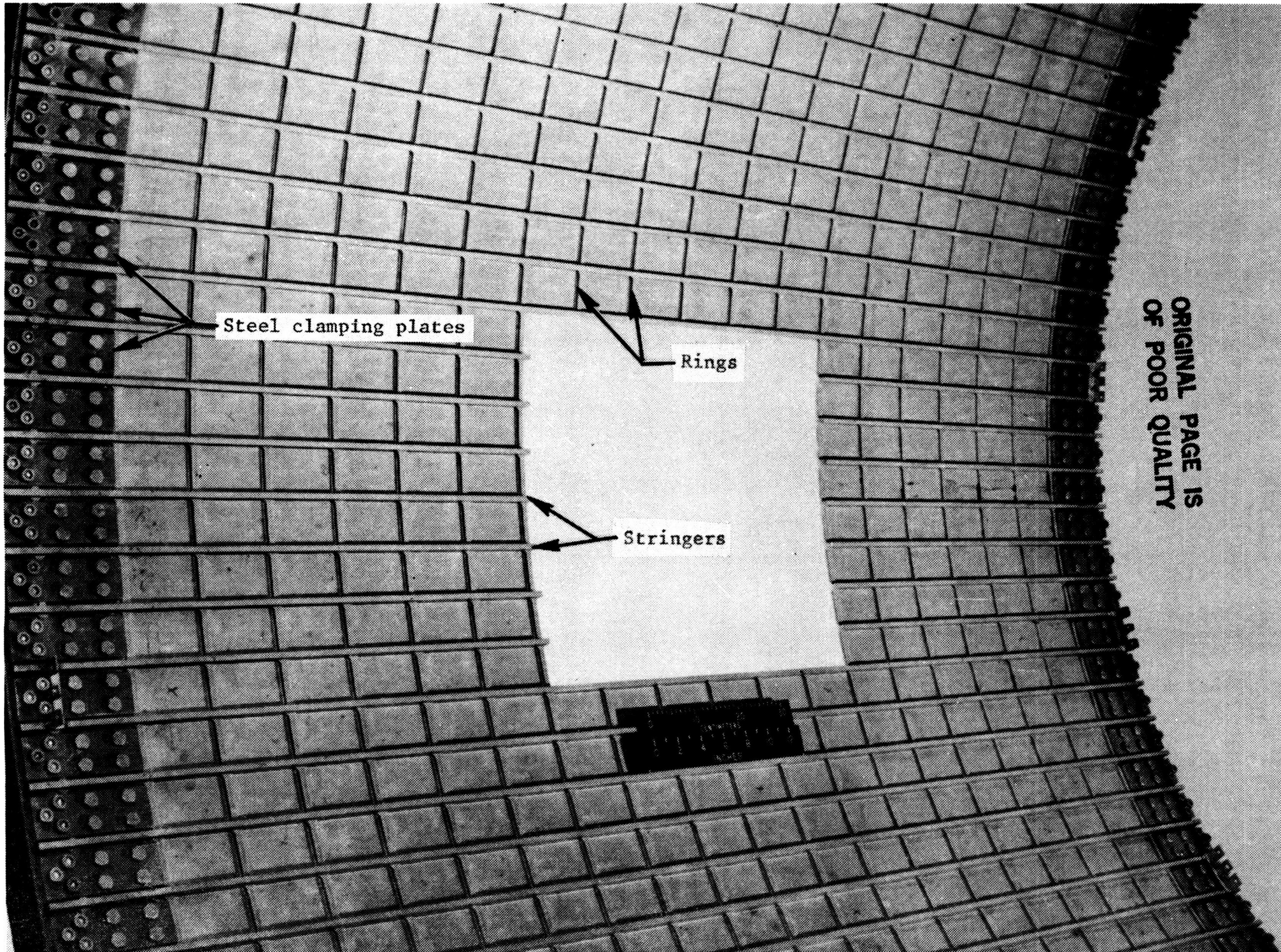


Figure 3.- Typical position of the edge of the cutout relative to rings and stringers (24x27 inch cutout shown).

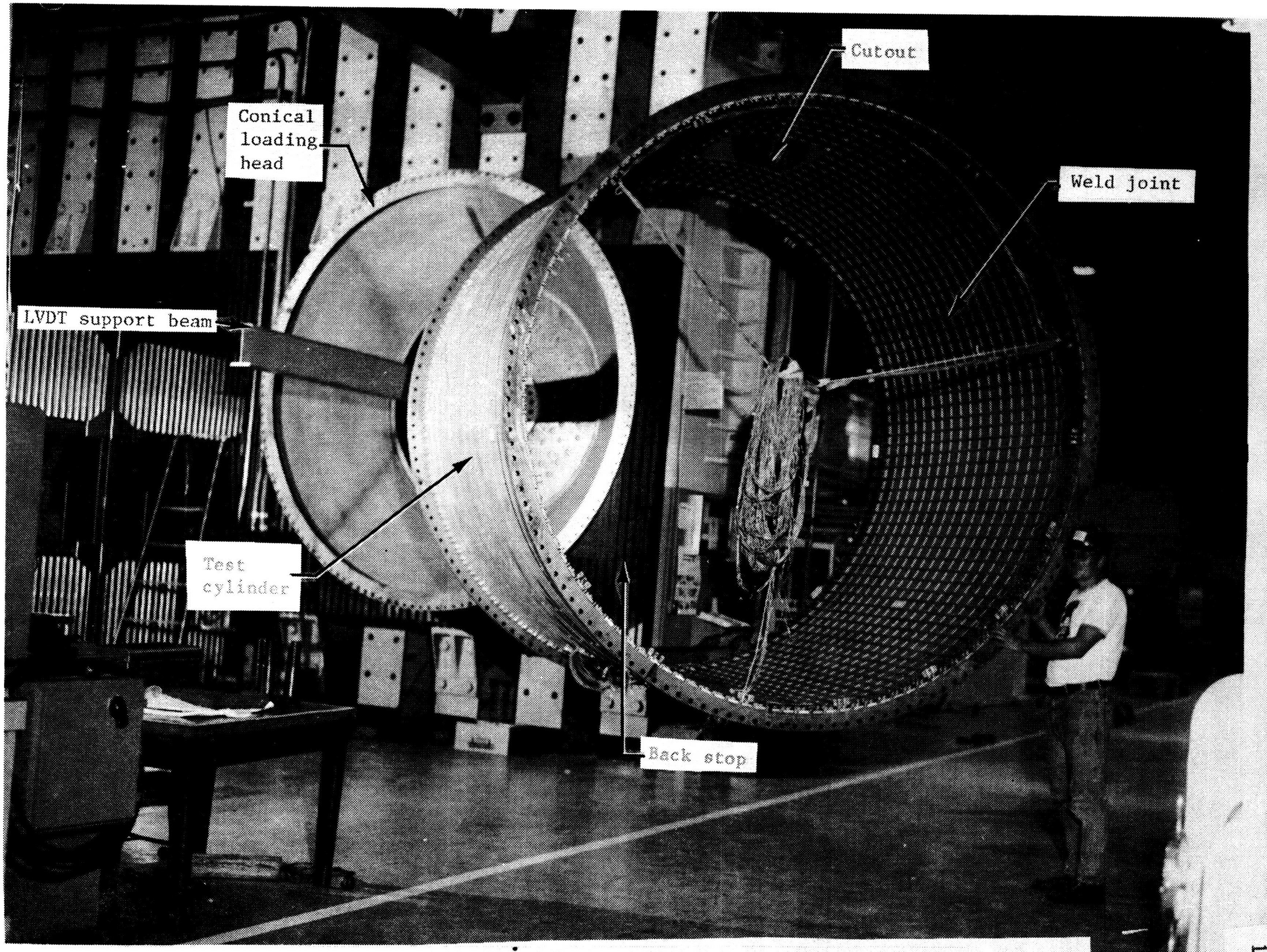


Figure 4.- Test shell being positioned in fixture showing LVDT support beam.

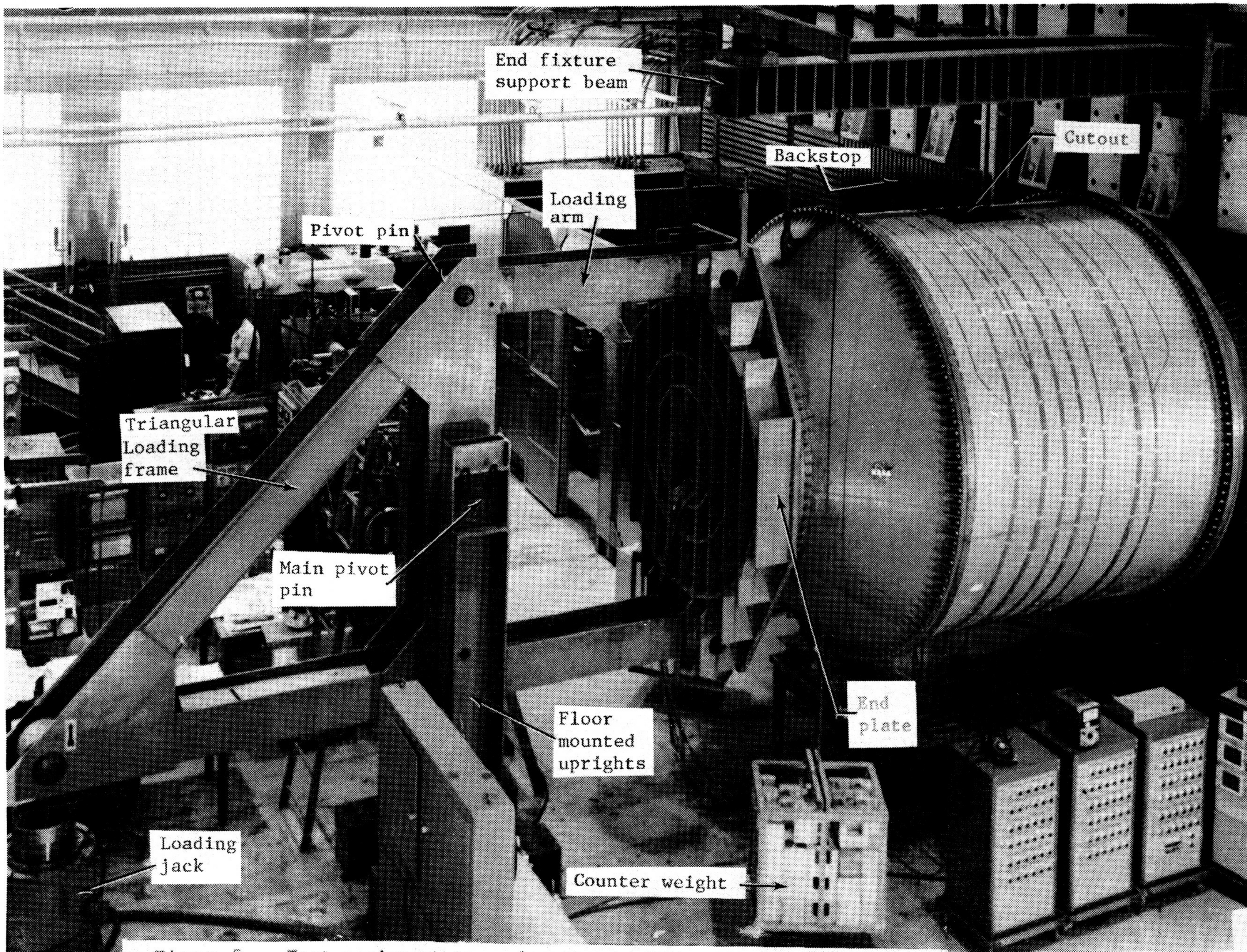
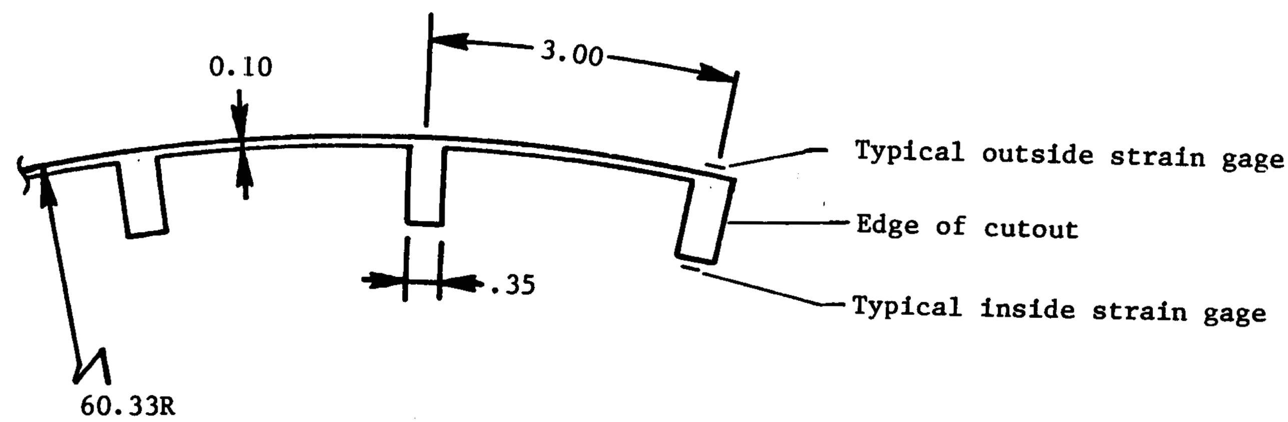
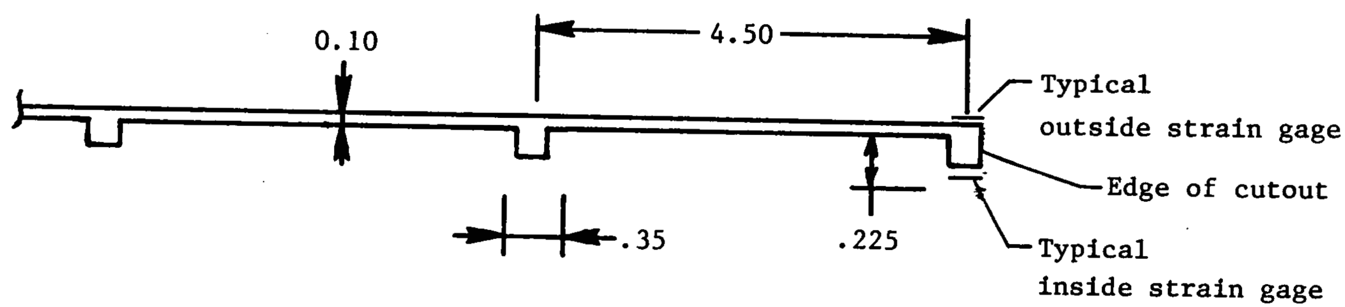


Figure 5.- Test equipment to apply an end bending moment to the shell.



(a) Typical location of back to back longitudinal strain gages on stringers at the edge of the cutouts



(b) Typical location of back to back circumferential strain gages on rings at the edge of the cutouts

Figure 6.- Location of strain gages placed around the edges of the cutouts.

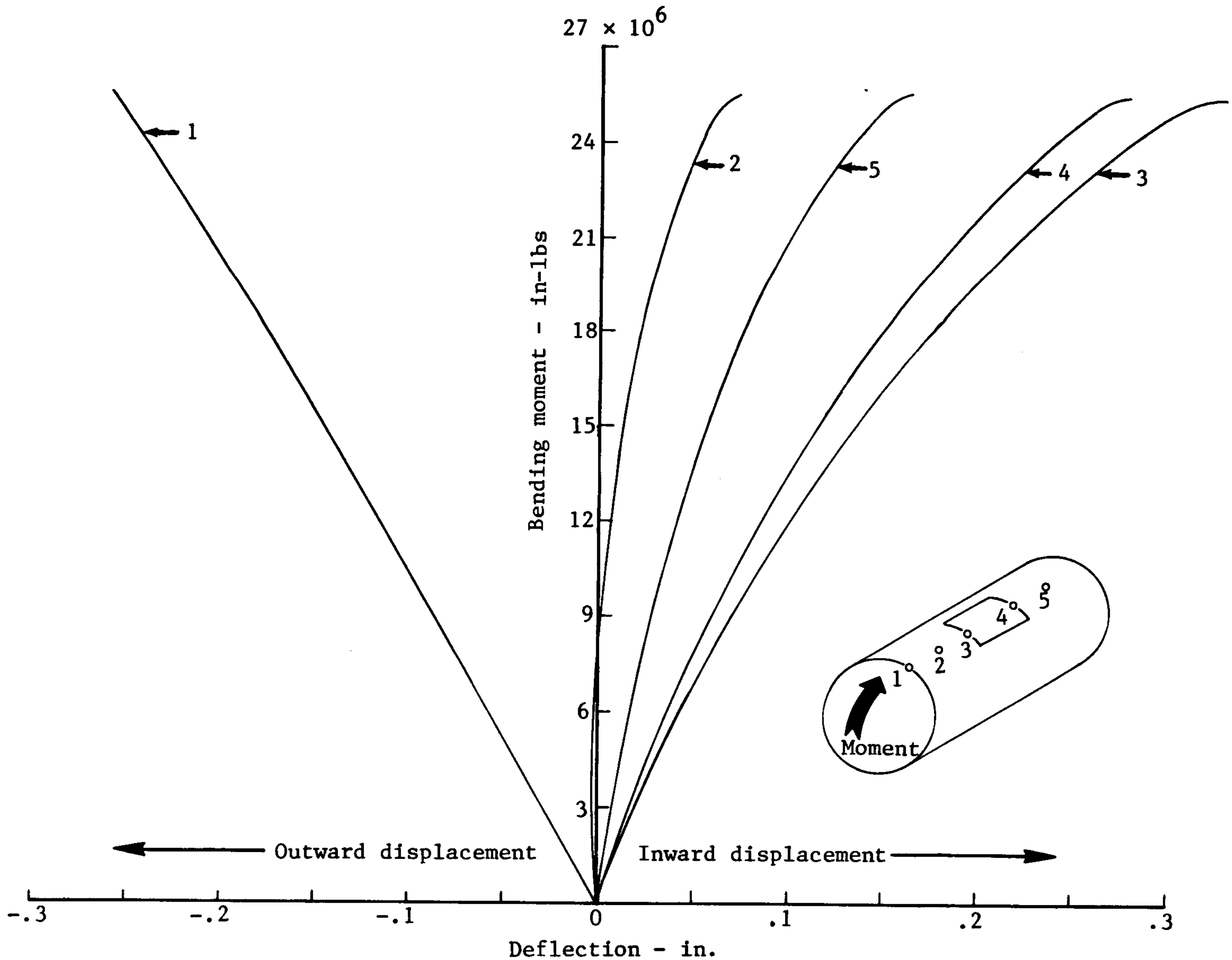


Figure 7.- Deflection on top centerline for 18x18 inch cutout.

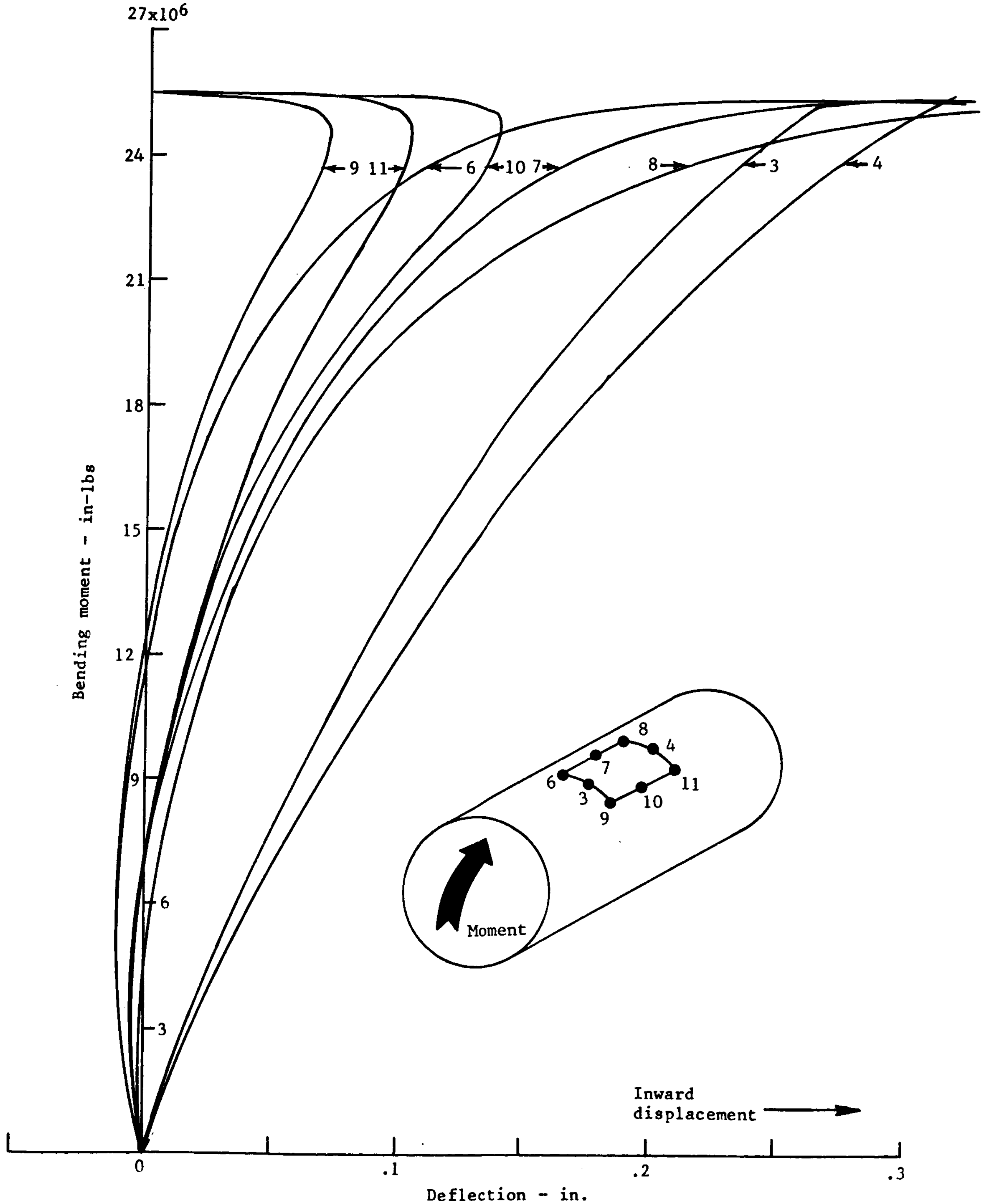


Figure 8.- Deflections around edge of 18x18 inch cutout.

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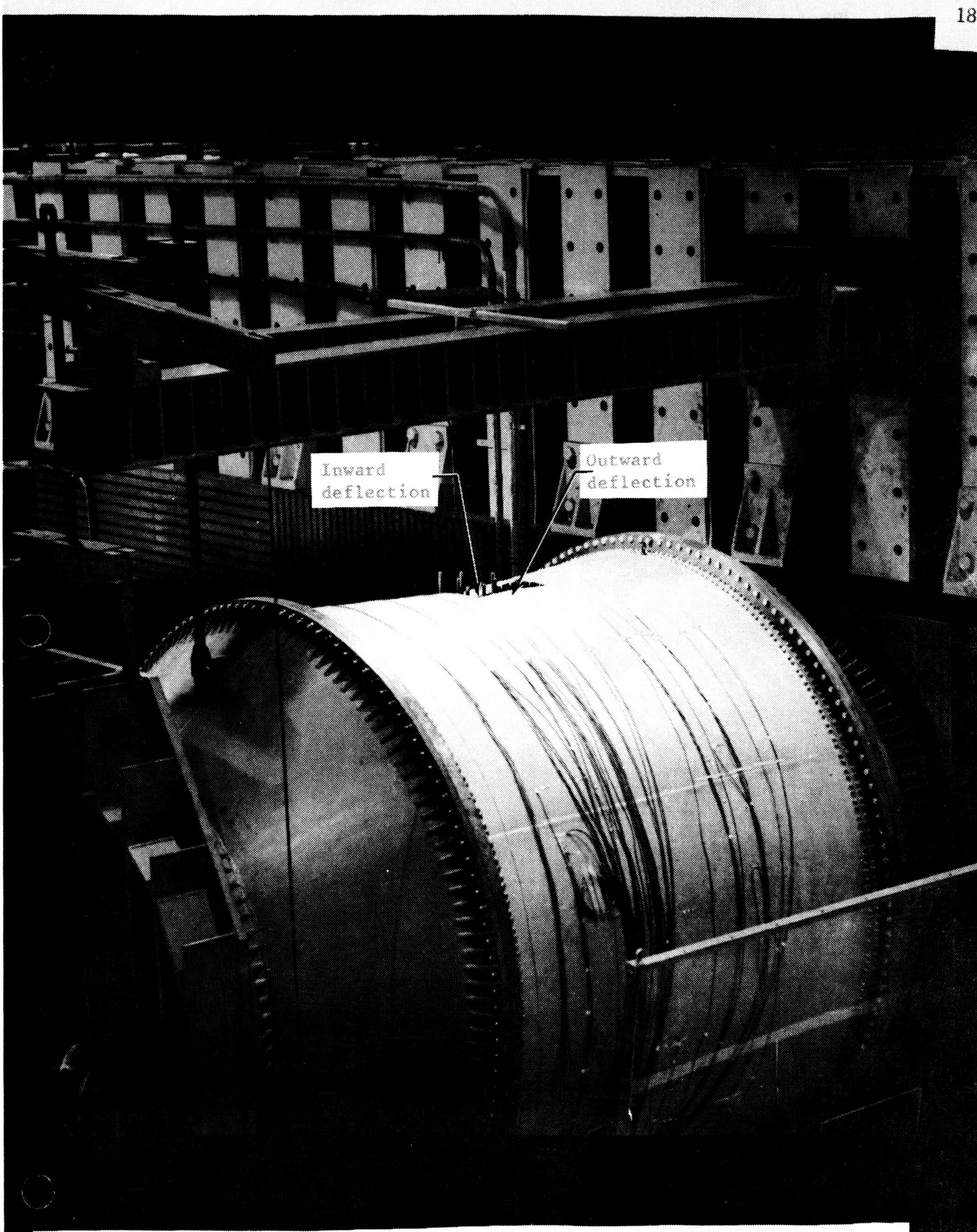

Figure 9.- Photograph of buckled cylinder showing buckled mode shape around 18x18 inch cutout.

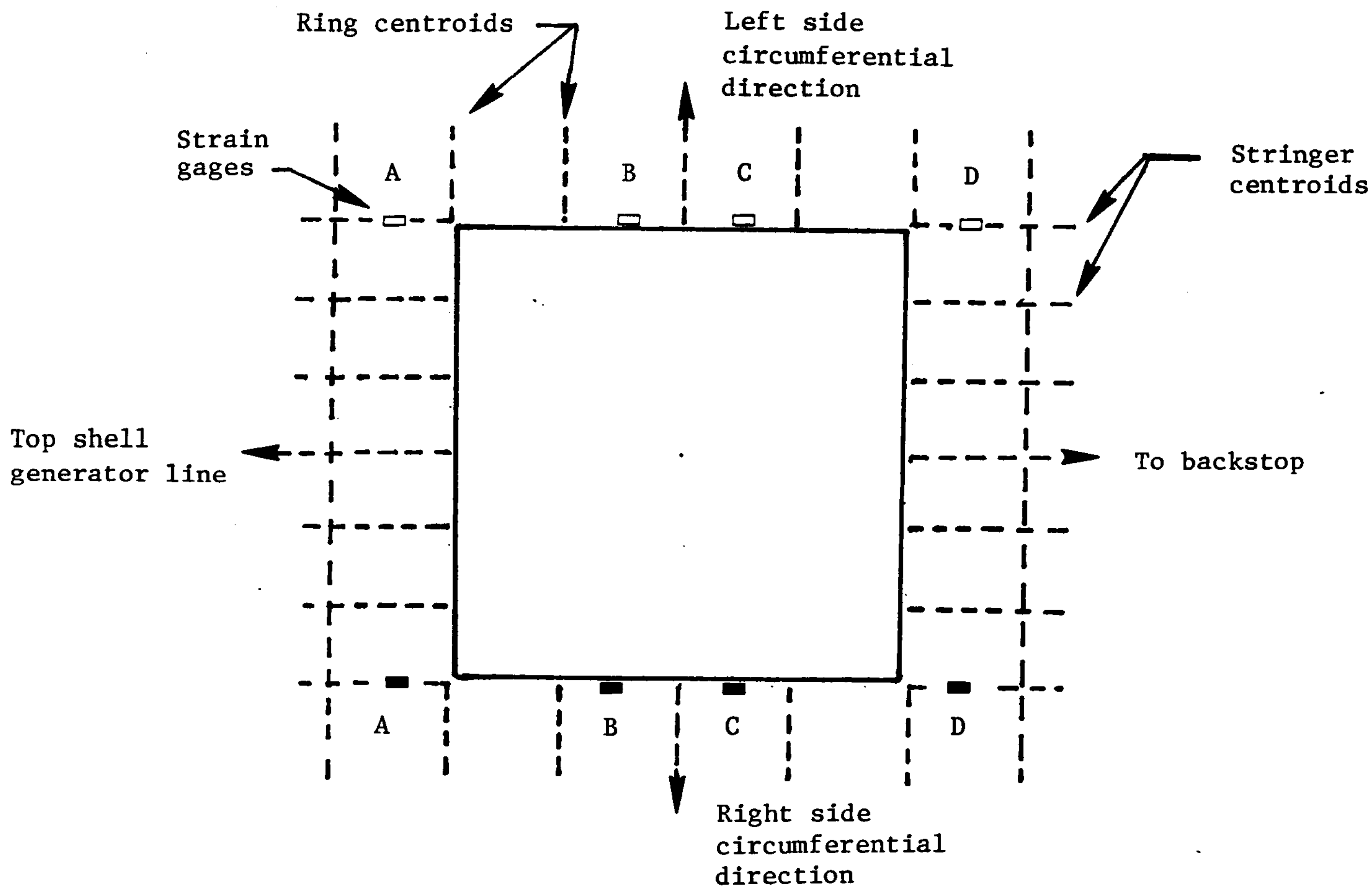

Figure 10.- Locations for longitudinal strain gages along right and left sides of 18x18 inch cutout.

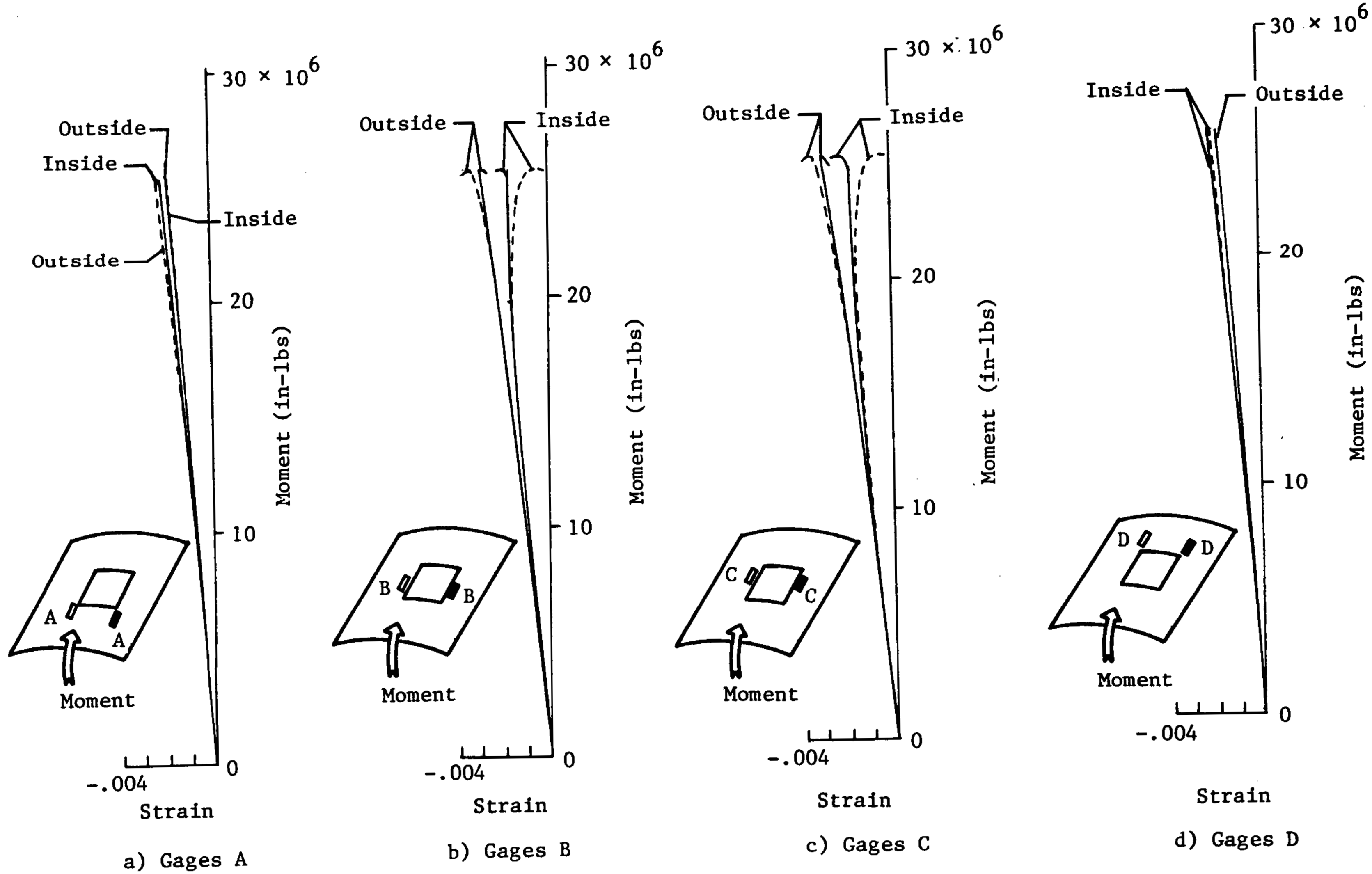

Figure 11.- Longitudinal strains on the right and left sides of the 18x18 inch cutout.

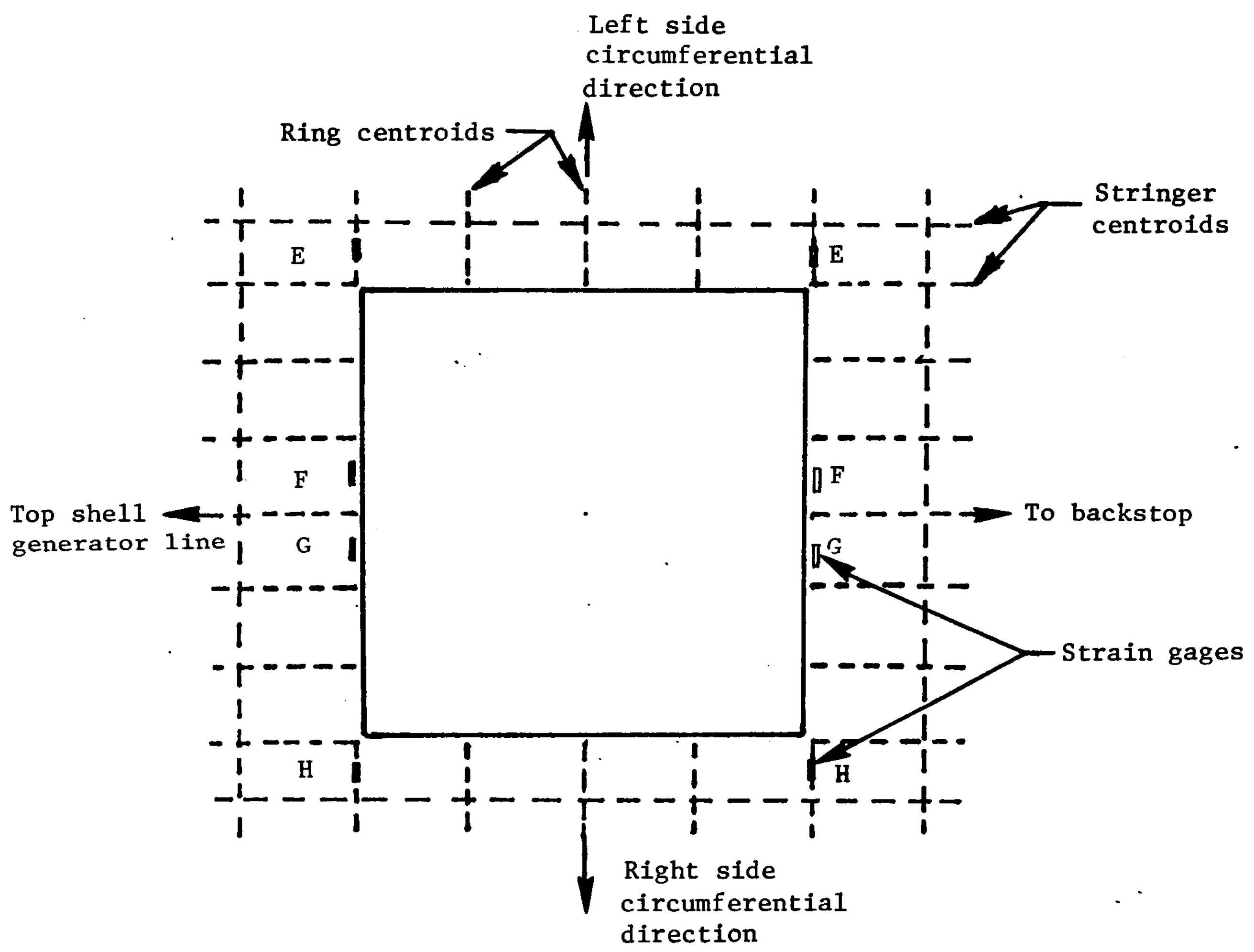

Figure 12.- Location of circumferential strain gages along front and back edges of 18x18 inch cutout.

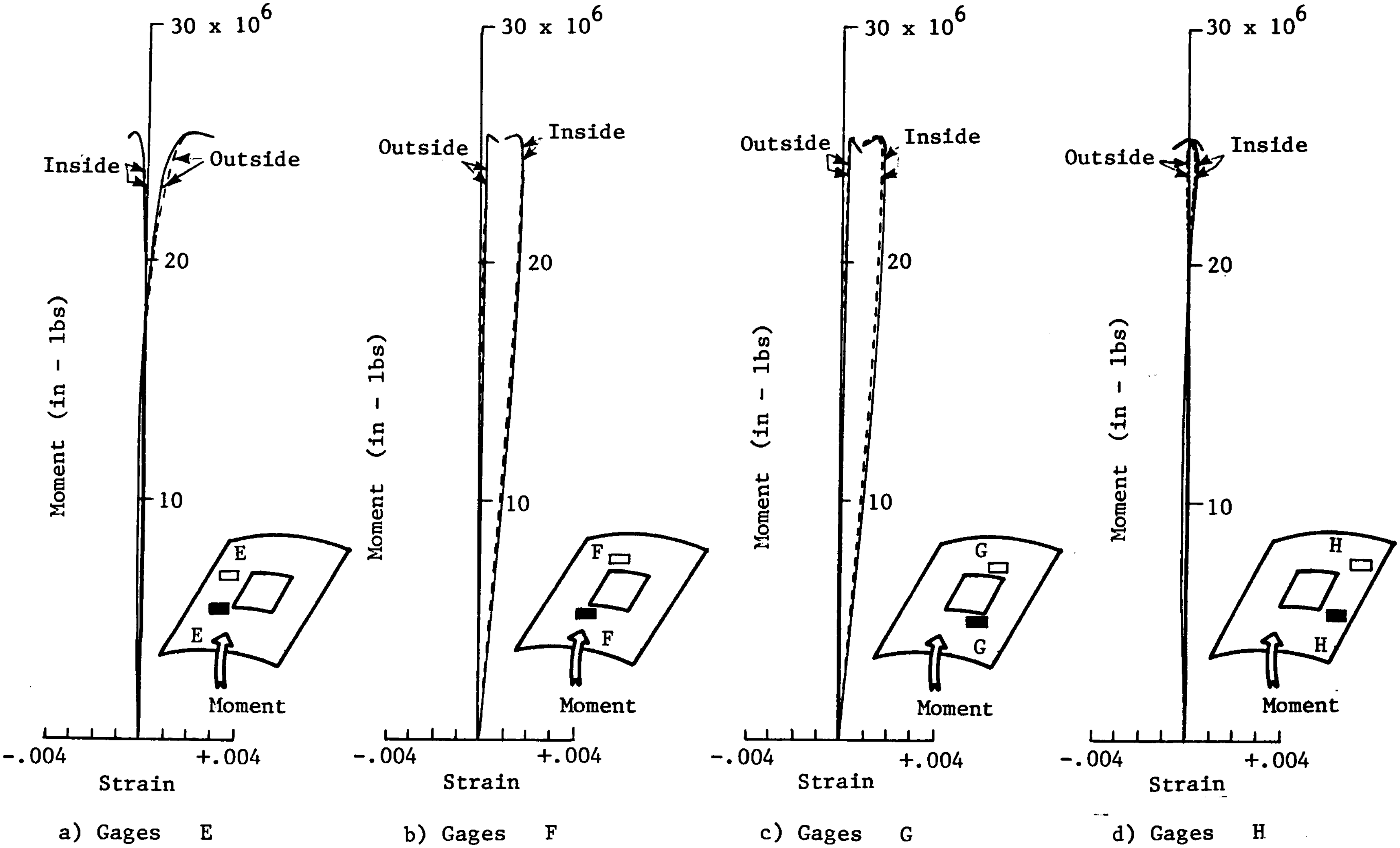

Figure 13.- Circumferential strains along front and back edges of 18x18 inch cutout.

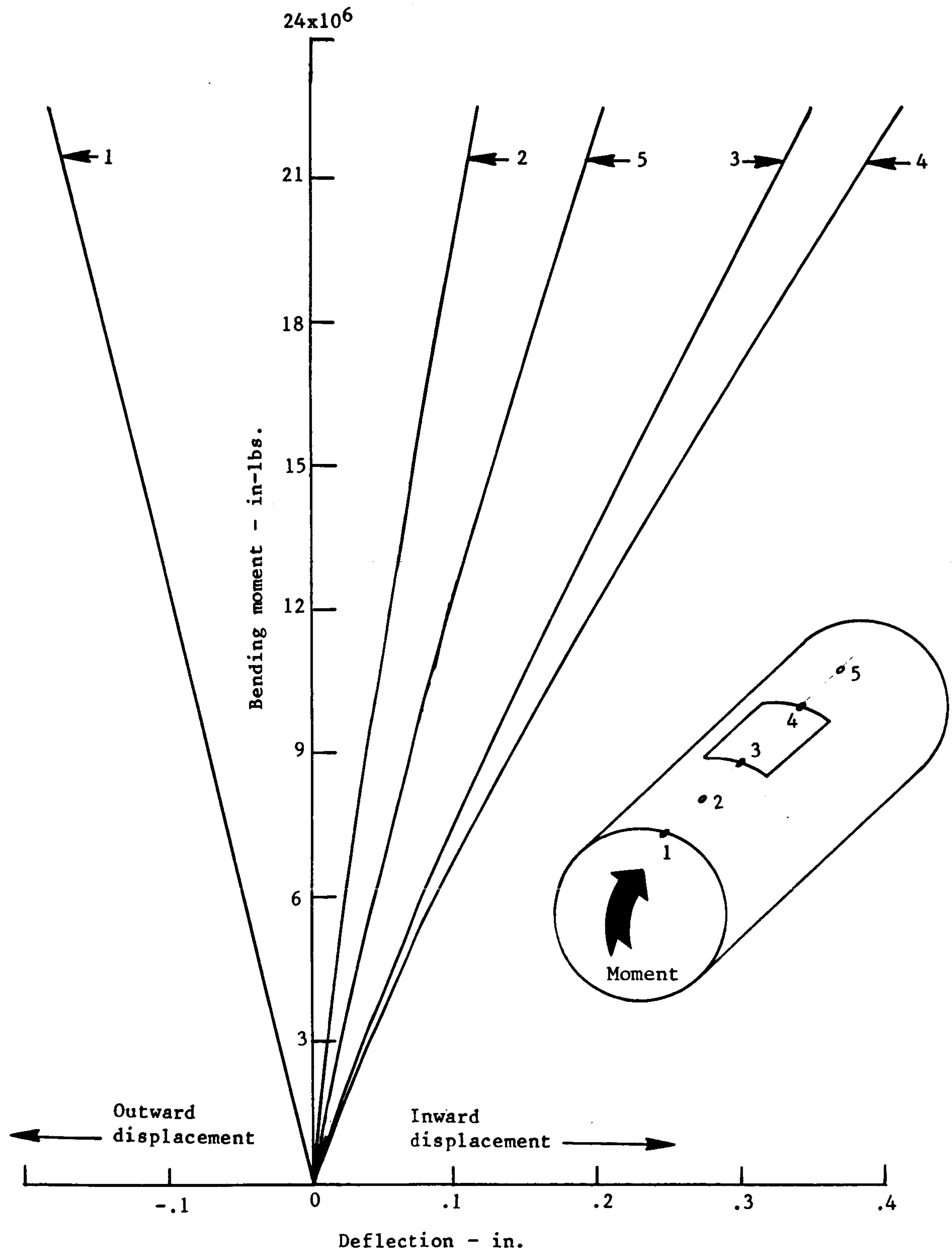

Figure 14.- Deflections along top centerline for 24x27 inch cutout.

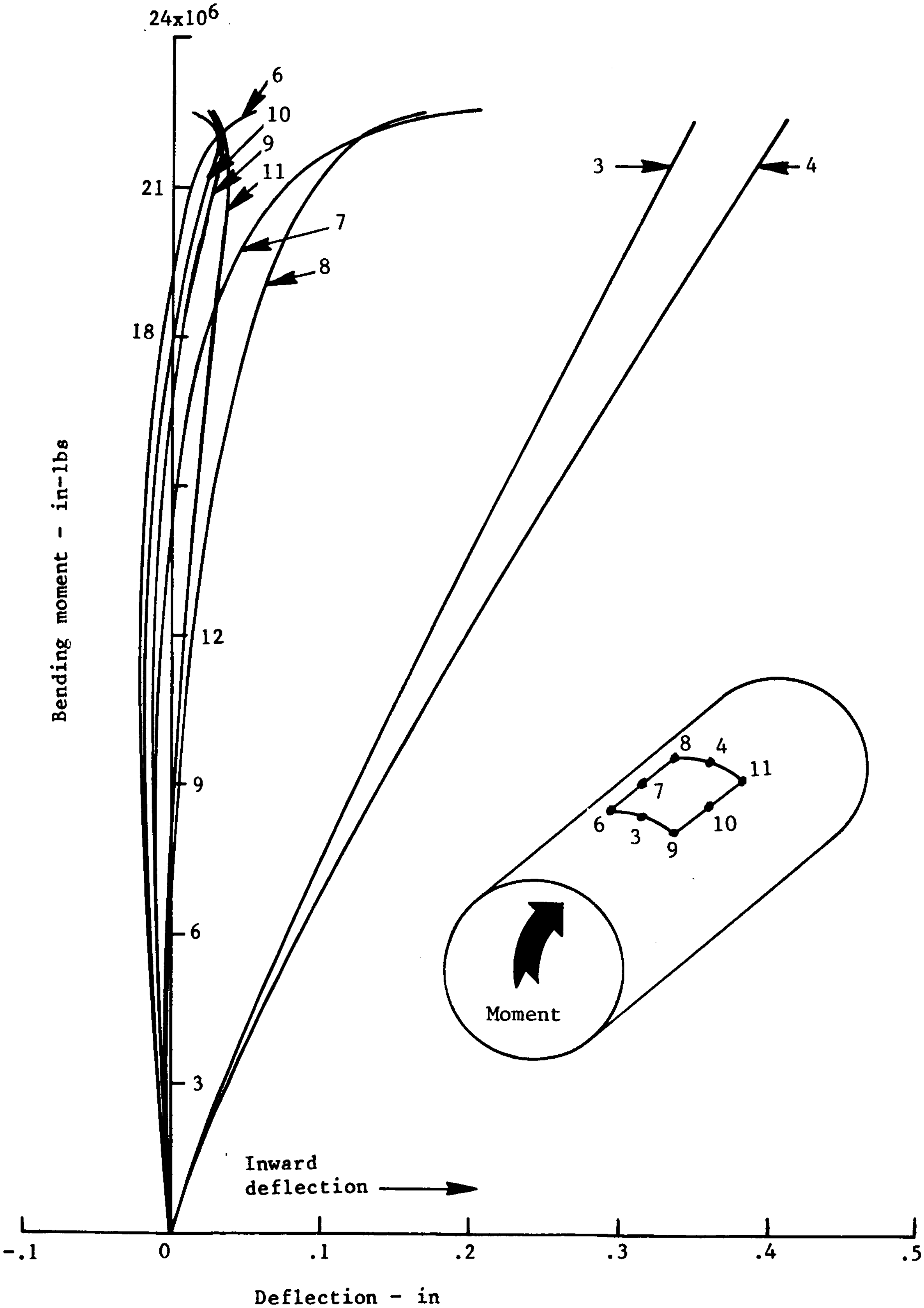

Figure 15.- Deflections around edge of 24x27 inch cutout.

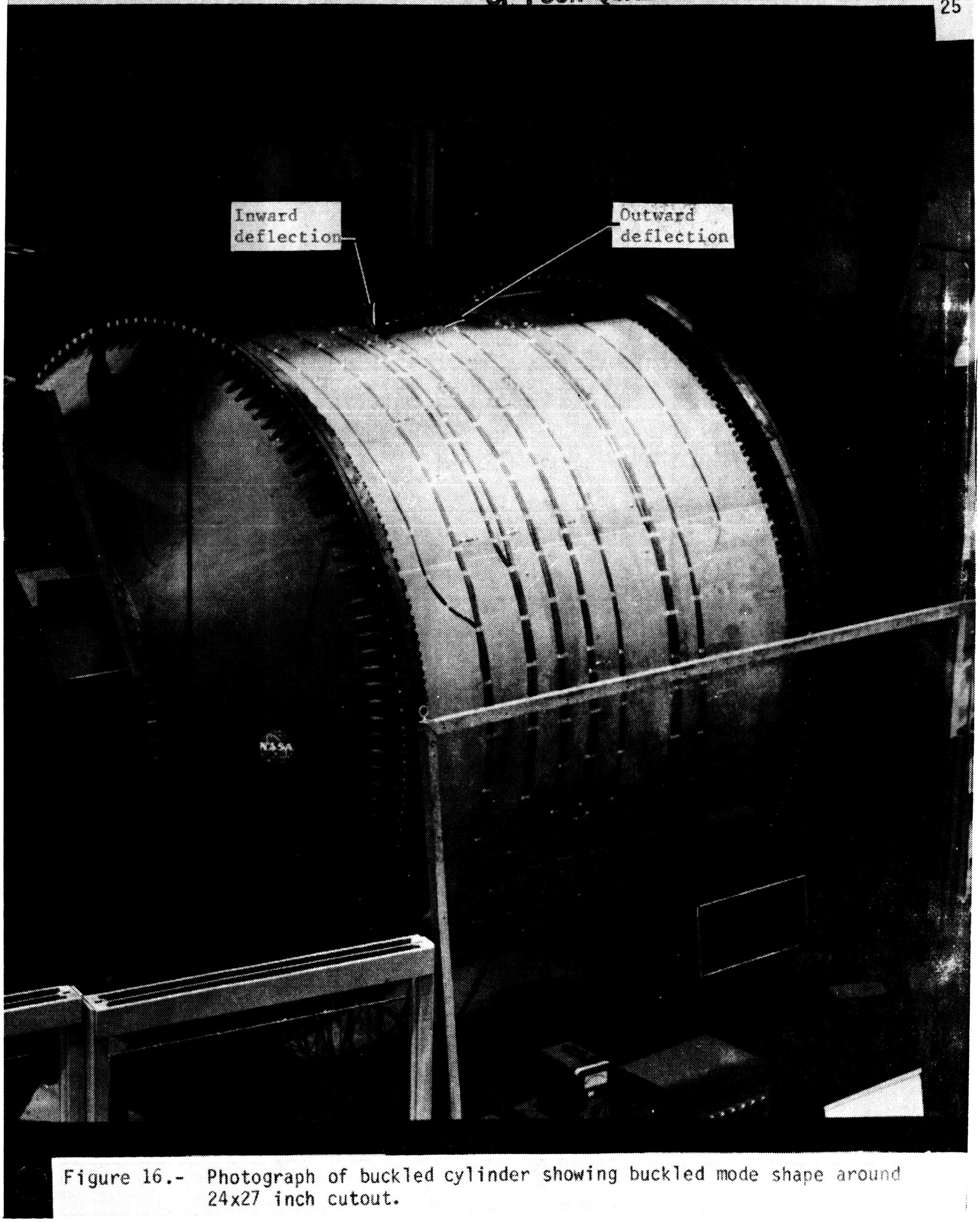

Figure 16.- Photograph of buckled cylinder showing buckled mode shape around 24x27 inch cutout.

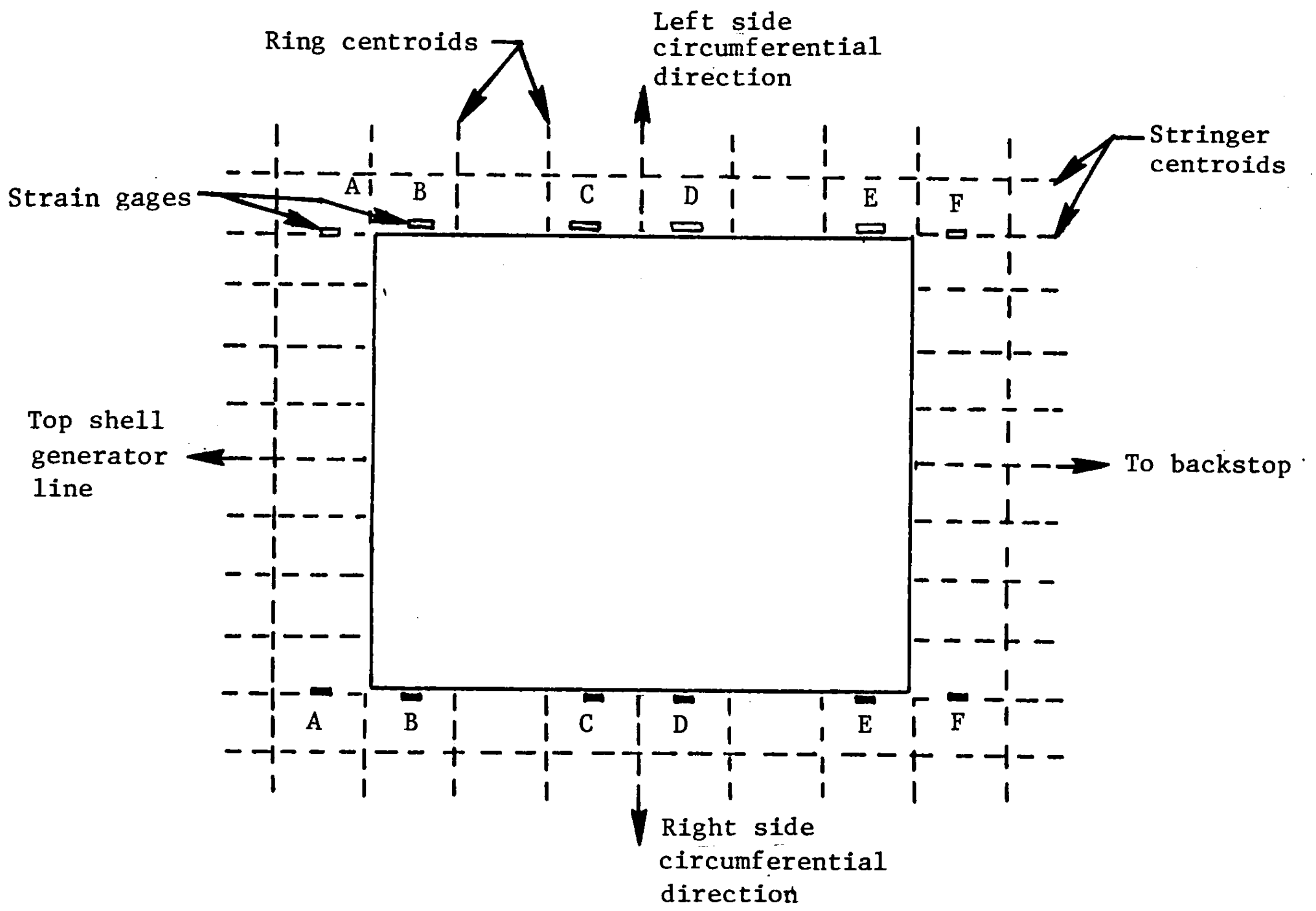

Figure 17.- Locations of longitudinal strain gages along right and left sides of 24x27 inch cutout.

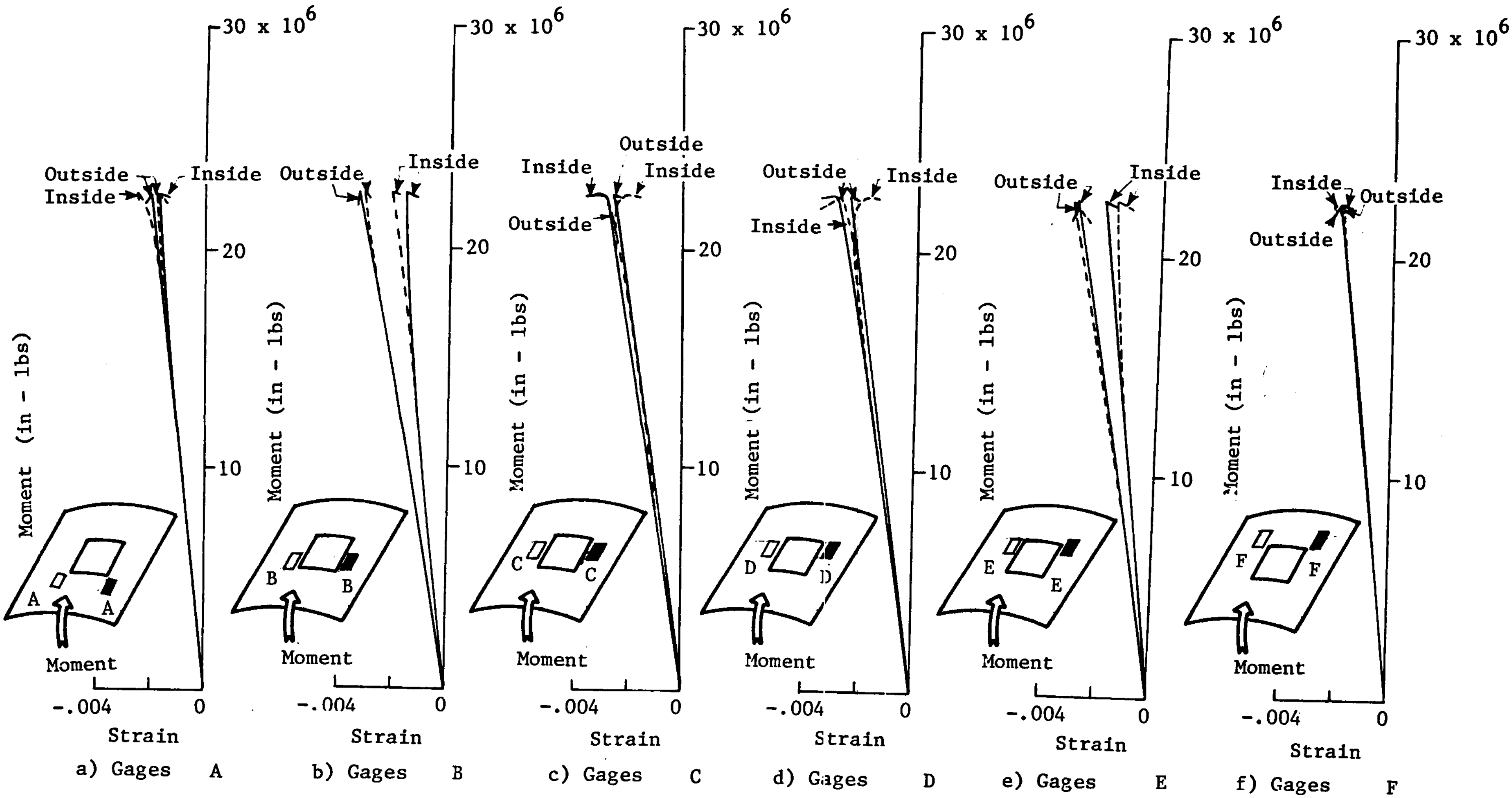

Figure 18.- Longitudinal strain along right and left side of 24x27 inch cutout.

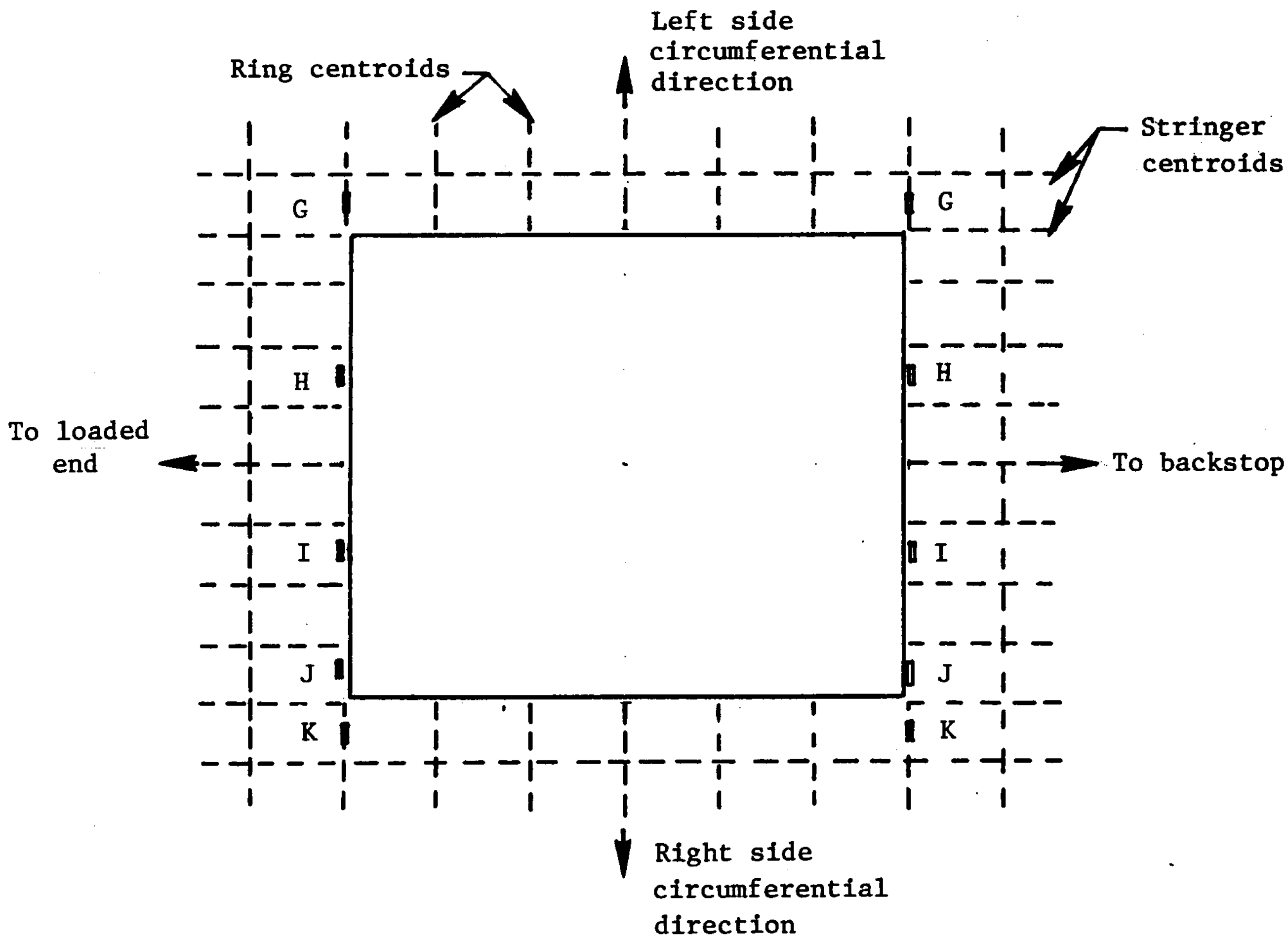

Figure 19.- Location of circumferential strain gages along front and back edges of 24x27 inch cutout.

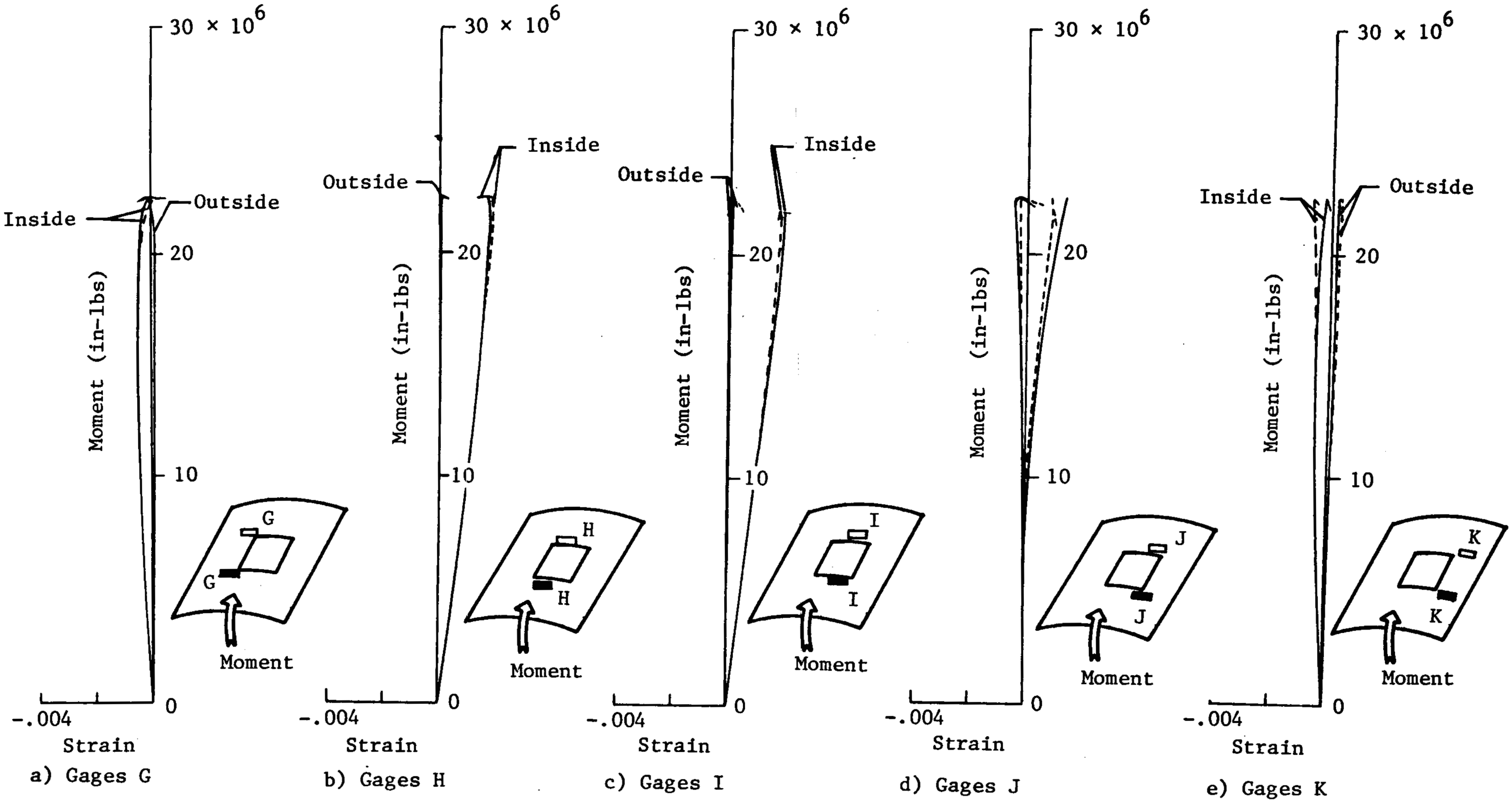

Figure 20.- Circumferential strains along front and back edges of 24x27 inch cutout.

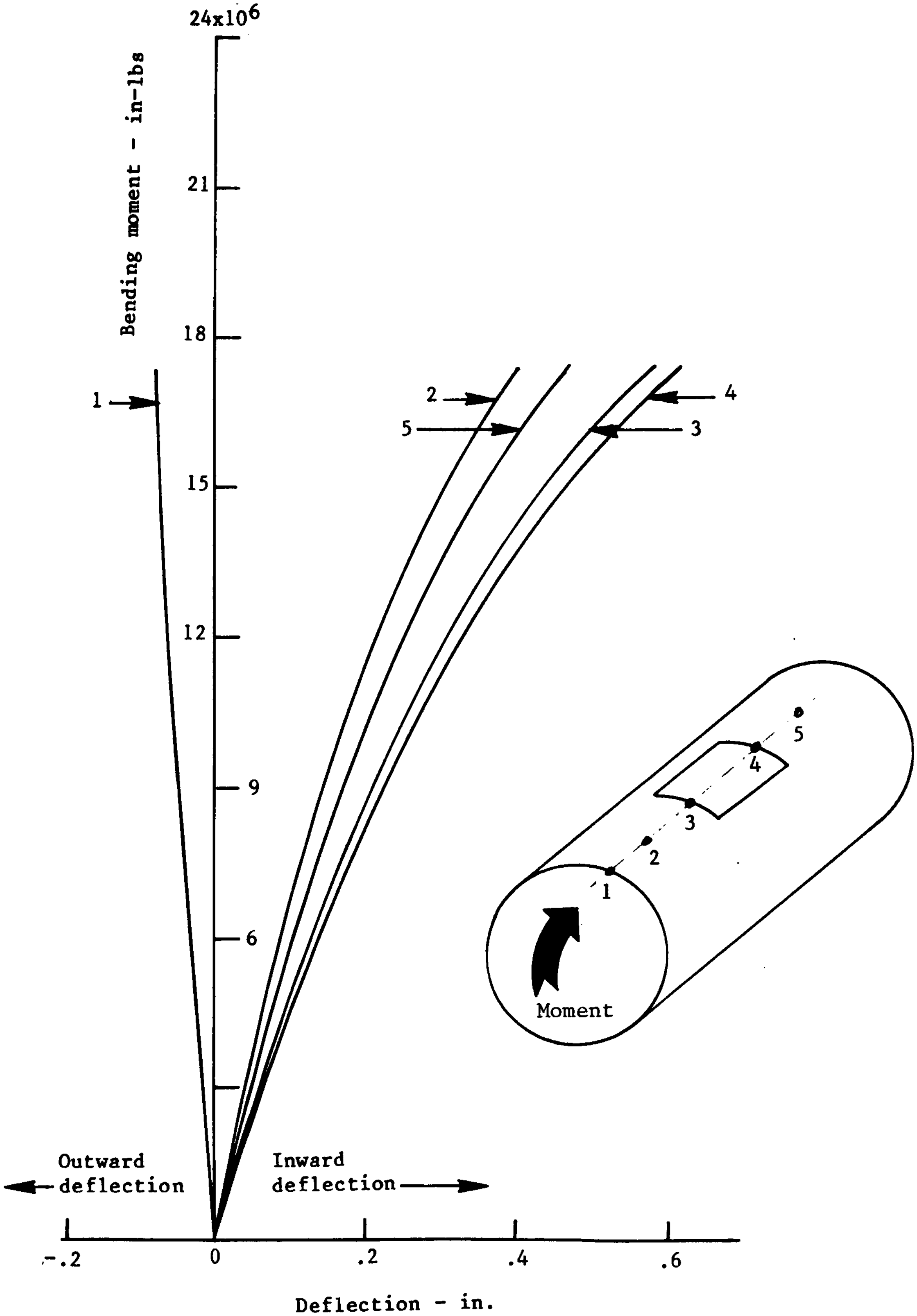

Figure 21.- Deflections along top centerline for 36x36 inch cutout.

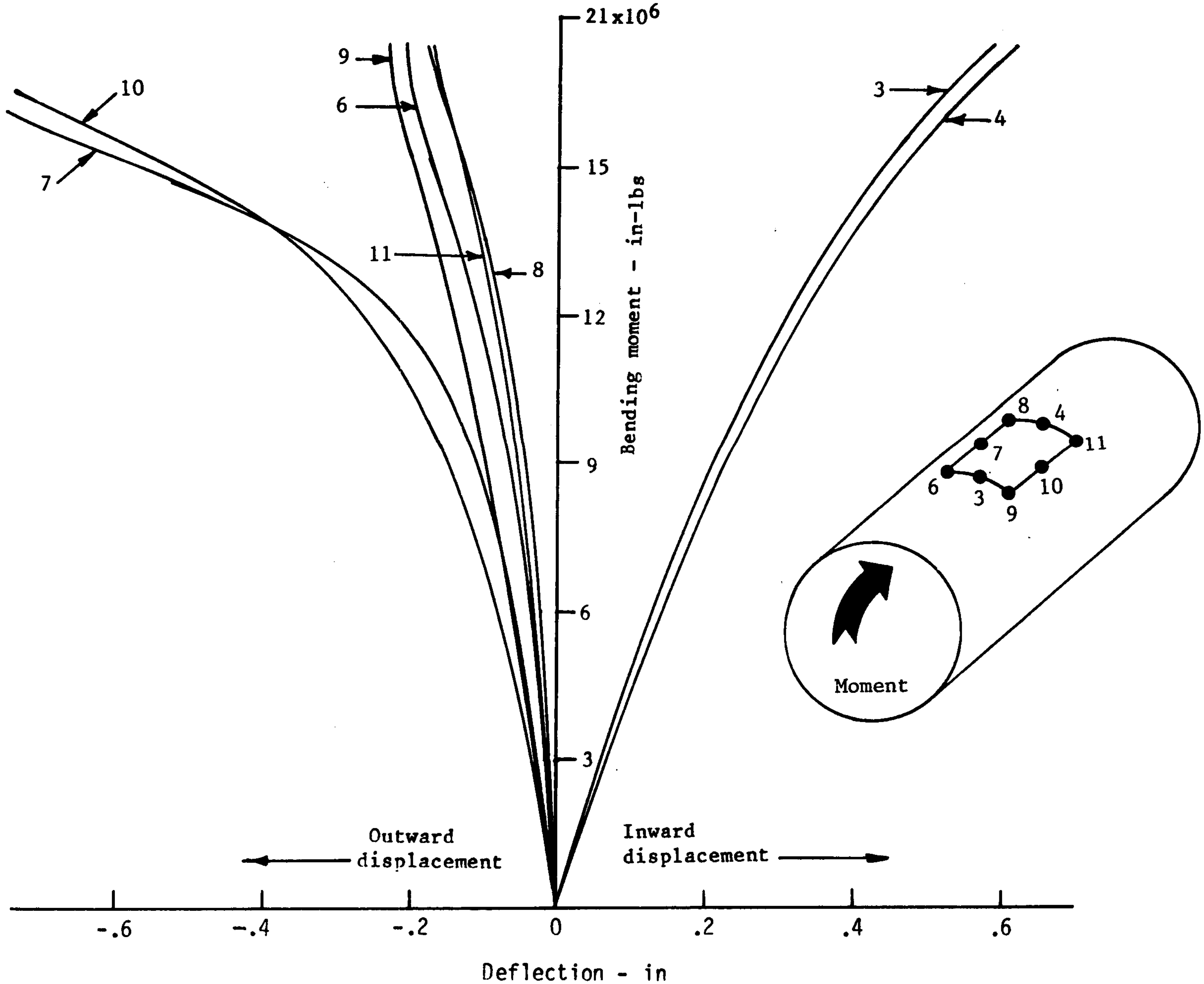

Figure 22.- Deflections around edge of 36x36 inch cutout.

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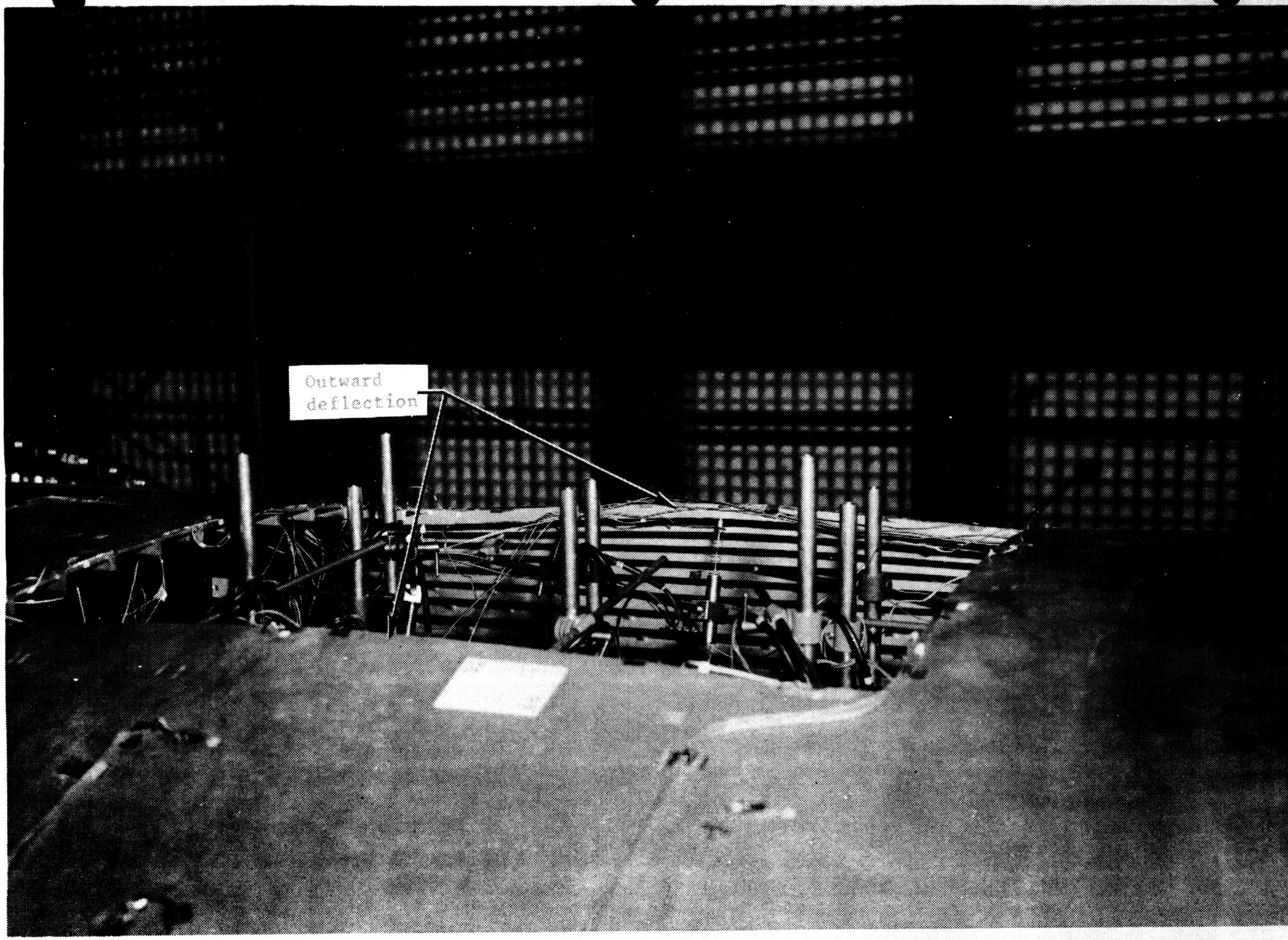

Figure 23.- Photograph of buckled cylinder showing buckled mode shape around 36x36 inch cutout.

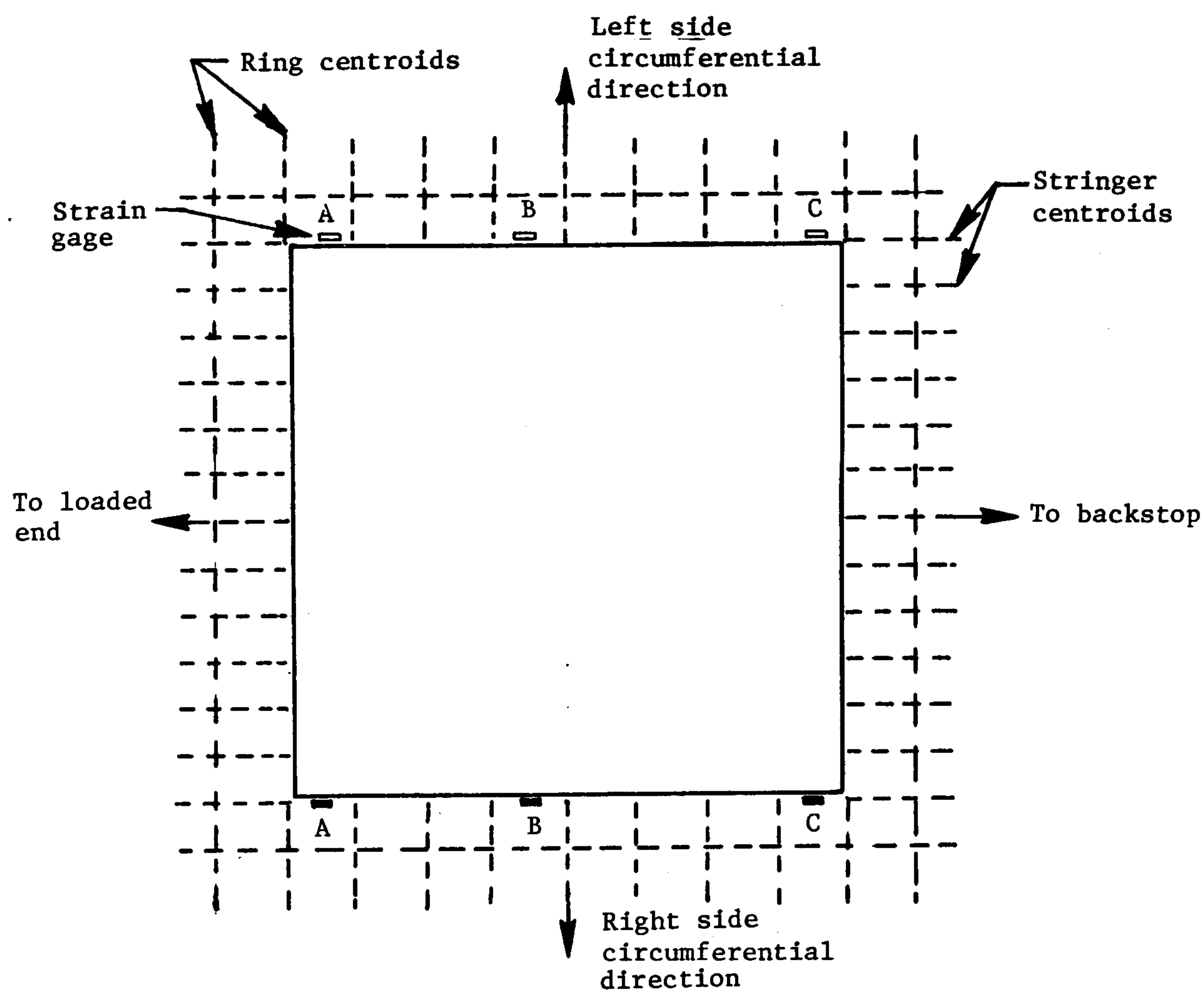

Figure 24.- Location of longitudinal strain gages along left and right sides of 36x36 inch cutout.

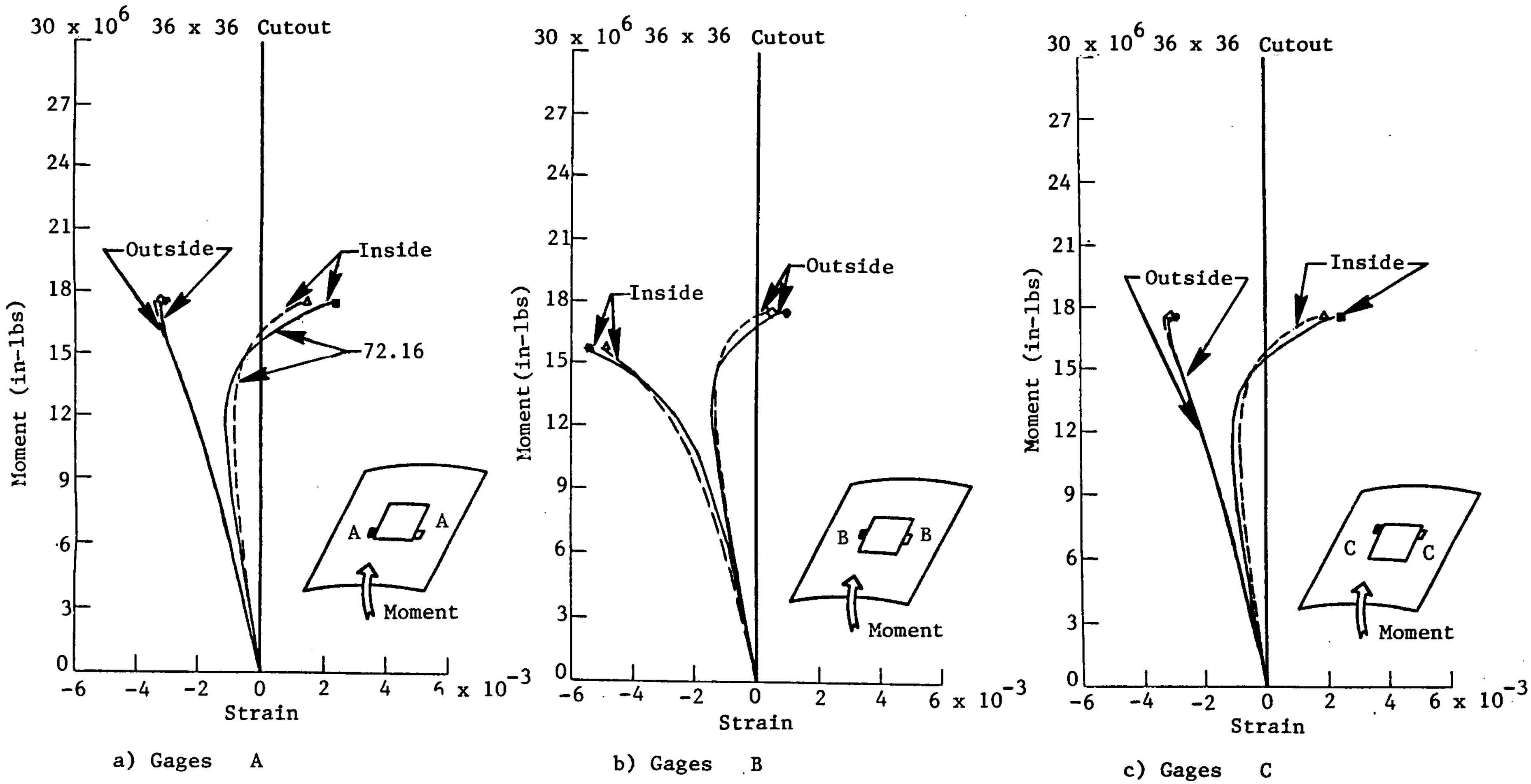

Figure 25.- Longitudinal strains along left and right sides of 36x36 inch cutout.

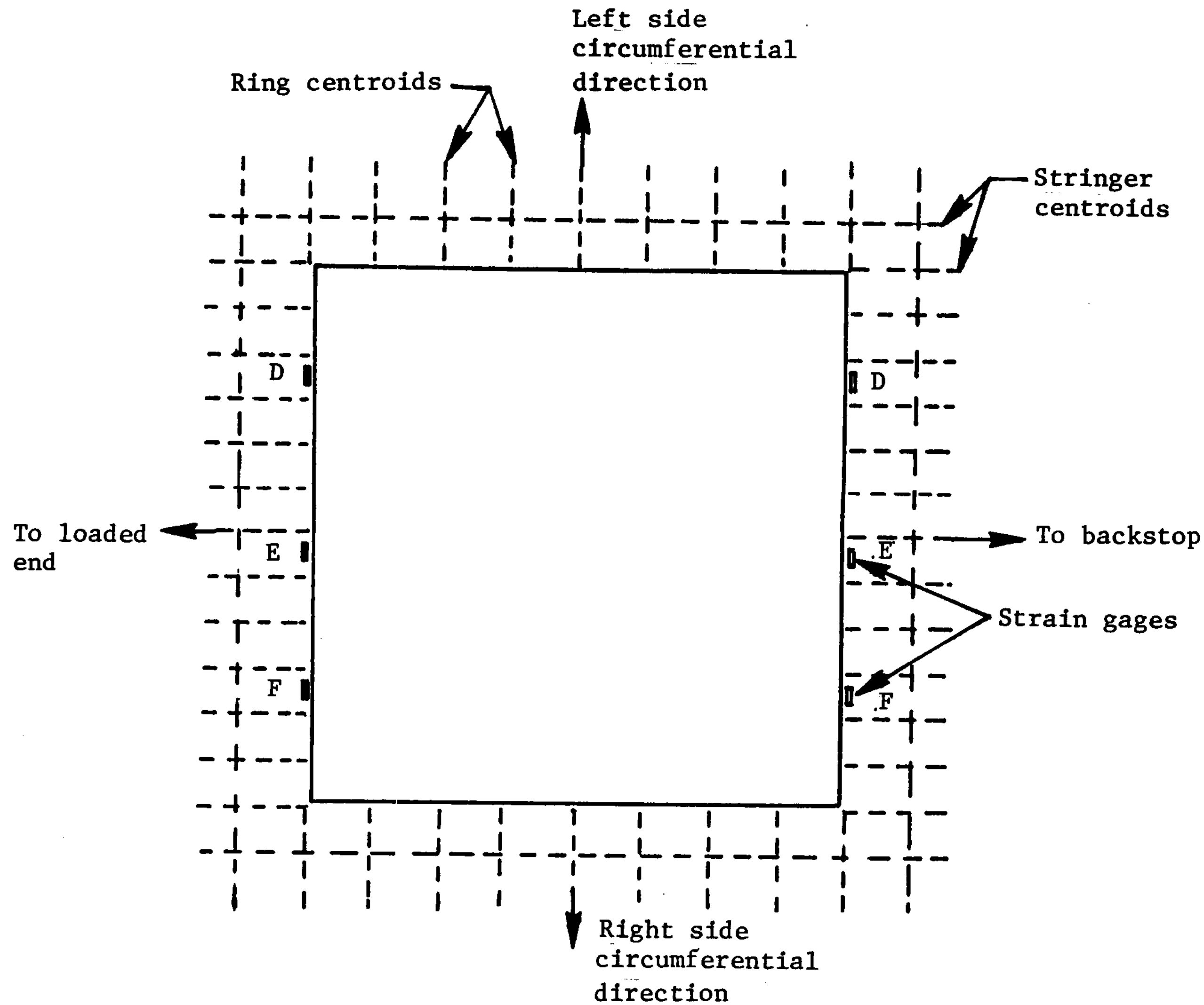

Figure 26.- Location of circumferential strain gages along front and back edges of 36x36 inch cutout.

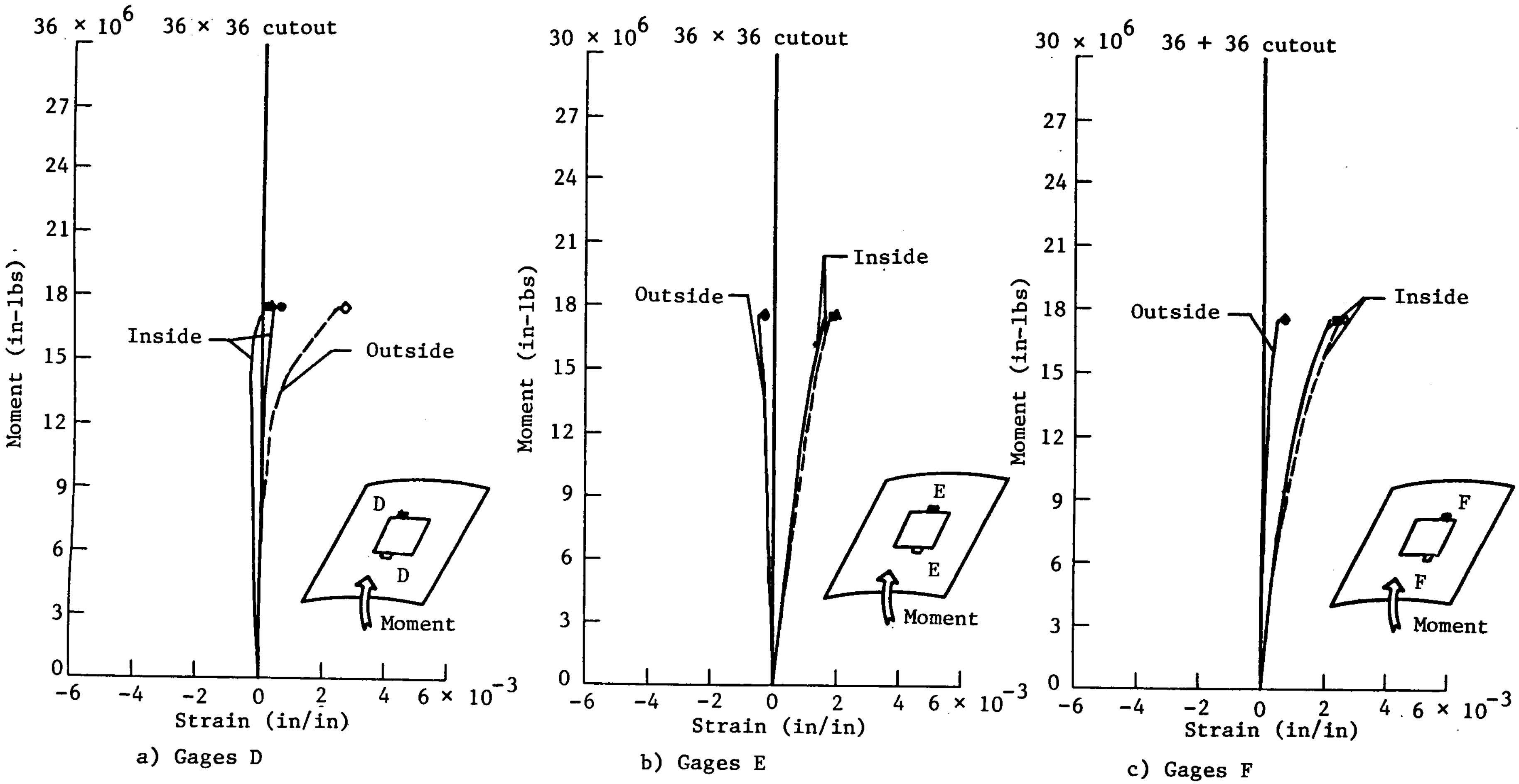

Figure 27.- Circumferential strains along front and back edges of 36x36 inch cutout.

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16. Abstract

A 10-foot diameter aluminum cylinder with rectangular cutouts in its ring and stringer stiffened wall was loaded to failure by an end bending moment. A 24x27 inch cutout, centered on the compression side of the shell, was first cut into the cylinder. After testing, the cutout area was enlarged to a 36x36 inch square cutout that removed the material damaged by buckling during the first test. After the second buckling test, the cutout area was patched with an equivalent stiffness plate bolted over the cutout hole. The cylinder was rotated 120° and a 18x18 inch square hole cut into the center of the new compression side of the shell. Test specimen details, test procedures, and test results are presented.

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